A TEXT-BOOK

OF

Mechanical Engineering

BY

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PART I.—WORKSHOP PRACTICE.

PART II.—THEORY AND EXAMPLES.

ELEVENTH EDITION, REVISED.

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PREFACE.

IT is now many years since the initiative of the City and Guilds of London Institute, in providing an examination for Mechanical Engineers, first suggested to me the desirability of writing the present text-book. In preparing students for this examination, I was being constantly asked for a comprehensive work which would at least show them the general lines on which their study, as engineer apprentices, should proceed ; and, in seeking to meet their request, I had to consider seriously (1) whether the whole theory and practice of Mechanical Engineering, or even a *précis* of it, could be compressed into one volume, and (2) whether it was desirable so to compress it. That this work has here been written is sufficient evidence of my own solution of the above questions—a solution which has been fully confirmed by the successful use, in teaching engineer students, of my chapters during the years of their preparation. I am perfectly aware that there are many who will object to any attempt to convey the *rationale* of practical processes by description on paper ; others may accuse me of 'cramming,' by attempting to condense the theories of engineering into half a volume. I would earnestly ask all these gentlemen, before condemning what may seem to them a too ambitious undertaking, to first consider carefully the following reasons which appeared to me to support my decision :—(1) The saving of time to the student, who need not now be always 'beginning at the

beginning,' the disadvantage of having to use a
small text-books; (2) saving of space, referenc
made when necessary to previous pages, obviatin
needless repetition: and here it may also be noti
in a single volume, embracing so vast a subject, 'p
had of necessity to be *nil*; (3) the examples of gr
successful writers to wit, Rankine, Ganot, Deschar
others; (4) the fact that practical processes are no
stantly described, with good result, both in the engi
journals and in the City Guilds' examination answe

I shall now explain the order of the chapters a
reasons for their arrangement.

Part I. makes the tour of the shops, with the in
of initiating the student into their mysteries. Th
method is to first describe the tools, after that the
processes, and then a series of graduated examples
application. A separate chapter is reserved for M
Tools. In the chapter on Fitting and Erecting, I
much difficulty in selecting suitable examples, th
being greatly interwoven. I therefore decided on t
of describing the construction of a horizontal engin
including most of the principal difficulties. Simila
the Boiler chapter, I have considered in detail the
out and building of a Marine Boiler. I am consci
Part I. is far from exhaustive, but the general met
first taking the castings and forgings, and then fol
the work through the shops to its completion, seem
proper course to pursue, and I hope will be endorsed

In Part II. I have treated, I believe, of all the pr
theories and investigations required by the engin
student. Some one has said that, when designin
engineer uses about one part of calculation to six of
ment. No amount of book study can impart the
most necessary quality: nothing but years of drawing
practice can effectively supply it; but any book sho

welcomed which attempts to lay the demon of 'rule of thumb,' the autocrat of even my own apprentice days. To encourage exactitude and prevent one source of error in the application of formulæ, I commence with a 'synopsis of lettering,' and have here introduced what I believe to be much needed, the retention of a certain letter wherever possible solely for one purpose. Though this was not always entirely practicable, I yet venture to think that some improvement has been effected. It is unfortunate, for example, that f stands both for stress and acceleration ; but at least it need not be adopted both for tons and pounds. I have, therefore, employed it for tons only. Again, velocity per minute and per second are better separately distinguished, as in the text, by the letters V and v respectively.

While never introducing mathematics unnecessarily, I have stated all the 'steps' that space permitted in such mathematics as have been introduced, and the latter will be found of but an elementary character, involving only simple equations, fractions, and the use of tables of sines and logarithms. The substitution of graphic treatment for the higher mathematics in many cases will, I think, be appreciated by most students.

As regards the order of Part II., the Strength of Materials without doubt comes first, to be followed by Energy and Kinematics ; these all assist in the treatment of Prime Movers worked by gases or liquids. With the knowledge acquired from Part I. and his own experience in the workshop, supplemented by the theory of Part II., the student should be able to commence the study of original design, for he is now in acquaintance both with what theory directs and the workshop restricts.

Regarding illustrations, I commenced with the intention of admitting no highly-shaded perspective views, which, showing nothing of interior parts, are only calculated to

confuse the student. Elaborate drawings, of course, necessitated great labour on my part, as well as considerable co-operation from makers and the editors of engineering journals. Such aid has in every case been afforded most ungrudgingly, and in many cases has exceeded my most sanguine hopes, both as regards drawings and matter. The necessity of well-detailed, modern examples, has always been present to me, and I confidently believe that such have been supplied. In connection with these, I would ask the reader to unite with me in thanking the following firms and gentlemen who have so kindly helped :—

Messrs. the Britannia Company, Colchester.
„ George Booth & Co., Halifax.
„ Joshua Buckton & Co., Leeds.
Mr. John Cochrane, Barrhead, N.B.
Mons. Delamare-Deboutteville, Rouen.
Messrs. the East Ferry Road Engineering Works Company, Millwall.
The Editors of *Engineering*, London.
Messrs. Greenwood & Batley, Leeds.
„ Andrew Handyside & Co., Derby.
„ Hulse & Co., Manchester.
„ B. & S. Massey, Manchester.
„ Priestman Bros., Hull.
„ David Rollo & Sons, Liverpool.
„ Samuelson & Co., Banbury.
„ Selig, Sonnenthal & Co., London.
„ James Simpson & Co., Pimlico.
„ Smith, Beacock, and Tannett, Leeds.
„ Smith & Coventry, Manchester.
„ the Sturtevant Blower Company, Boston, U.S.A., and London.
„ Tangyes Limited, Birmingham.
Mr. Ralph H. Tweddell, Westminster.
Sir J. Whitworth & Co., Manchester.
Mr. Wilson Worsdell, of the N.E. Railway.

I have also to thank my assistants at the Goldsmiths' Institute, Mr. William Ashton and Mr. George T. White, for much valuable aid in the correction of proofs, and Mr. R. W. Weekes for assistance in the matter of electric transmission.

In conclusion, it is my sincere wish that the book may prove of real benefit to engineers of every class. In furtherance of this, I will gladly explain any portion that may seem abstruse, and shall be greatly obliged by having any errors pointed out. I must finally state that I do not intend the work to be merely an aid to any particular examination, but I have introduced whatever seemed to me most helpful to those for whom it has been prepared.

WILFRID J. LINEHAM.

Goldsmiths' Institute, New Cross, S.E.
October, 1894.

PREFACE TO THE SECOND EDITION.

THE demand for a Second Edition of this book occurring so soon, the First Edition having been exhausted within a few months of publication, I have only had opportunity to make the usual corrections, and to add the more necessary extensions in an Appendix.

Speaking generally, I am highly gratified at the reception which has been accorded to the work, especially considering the imperfections of its First Edition, and here take occasion to sincerely thank correspondents and reviewers both for praise and for help.

As there has been some little misunderstanding in the reading of my first preface, let me at once say that the only examination I had in view when preparing the book was that which is the best examination for the engineer, 'his actual requirements in doing the world's work.'

Once more I earnestly solicit corrections and suggestions, renewing at the same time my offer to explain any difficult portion.

WILFRID J. LINEHAM.

Goldsmiths' Institute, October, 1895.

PREFACE TO THE THIRD EDITIO[N]

IN this edition some corrections and considerabl[e] tions have been made, with the desire of satisfi[ ] filling gaps and keeping the work abreast of modern m[ ]

The drawing and description of Whitworth's Steel process, p. *790*, are taken from the proceeding[s] Inst. of Civil Engineers, with the kind permission Council, and of Mr. W. H. Greenwood, M.I.C.[E] author. I am similarly indebted to the Inst. of Mec[h] Engineers for a short account of the experiments [ ] Research Committee on Friction, p. 870. The ma[ ] Graphical Calculus and n^{th} Moments, pp. 846–7 and I owe to Prof. Karl Pearson, F.R.S.; and the expe[ ] on Machine Efficiency, pp. 873–4, on High-speed F[ ] p. *869*, and on Mechanical Hysteresis, p. 837, wer[e] at University College, London, with the kind assista[nce] supervision of Prof. T. Hudson Beare, B.Sc., M.I.C.E[.] formulæ for thick cylinders, applied to ordnance, ar[e] from the *Gunnery Manual*; and the investigation [ ] compression Efficiency, p. 880, though slightly cha[nged] originally due to Prof. W. W. F. Pullen, Wh. Sc, whom I received it. The drawing and description [ ] J. H. Gibson's Worm-wheel Cutting Machine, p. [ ] received from that gentleman; the liquid-fuel illust[ ] on p. 907, from Mr. R. Wallis, Wh.Sc.; and Mr [ ] Pretty, Wh. Sc., has supplied much first-hand wo[ ] information. Others have kindly corrected errors [ ] gested improvements. To these, and to all the [ ] mentioned authorities, I tender my most hearty than[ks]

If there is any borrowed matter of importan[ce] source of which I ought to have stated, I sha[ll be] grateful to have it pointed out to me.

WILFRID J. LINE[HAM]

Goldsmiths' Institute,
August, 1898.

PREFACE TO THE FOURTH EDITIO[N]

THIRTY-ONE pages have been added in the [form] of a Third Appendix, dealing with matte[rs that] seemed of importance; and the whole book has

carefully re-read and corrected. The excellent and ready method of treating redundant members given on pp. 924–5 is due to Mr. Max am Ende, though obtained from Prof. Pullen's work on *Graphic Methods*, and here further simplified. Other sources of information have been acknowledged in the text.

WILFRID J LINEHAM.

Goldsmiths' Institute,
January, 1900.

PREFACE TO THE FIFTH EDITION.

ANOTHER short Appendix has been added to bring the work up to date, and the general text has been thoroughly revised.

WILFRID J. LINEHAM.

Goldsmiths' Institute,
October, 1901

PREFACE TO THE SIXTH EDITION.

THE continued progress of the Science of Mechanical Engineering necessitates a further Appendix, the fifth; for a book of this kind must be kept up to date at all costs. We have been requested to publish the Appendices separately, but it is found impossible to do this without loss; so are constrained to advise readers to transfer their early editions to younger students, for whom they are still useful.

We have sometimes been asked to divide the book into separate volumes, but are just as often urged to keep it intact. In view of these conflicting requests, it appears advisable to retain the present form of a self-contained manual, the want of which was the original reason for writing the book.

It is impossible to specialise in such a work. The matter can only be taken as a general statement of science and practice; and types only can be used as examples. I trust that specialists will consider this point whenever they are apt to criticise a little harshly the chosen types.

Lastly, I would urge readers to make regular use of the Appendices, which are fully connected by reference notes

with the body of the book; and I have on
express my gratitude to correspondents and c
for their continued help.

WILFRID J. L

Goldsmiths' Institute,
January, 1903.

PREFACE TO THE SEVENTH EDI

I HAVE to thank the following firms for cc to the Sixth and Seventh Editions:—

Messrs. Bopp & Reuther, Mannheim.
„ Greenwood & Batley, Leeds.
„ Palfreyman, Liverpool.
„ Alfred Herbert, Limited, Coventry.
„ Pfeil & Co., London (acting for J. Reinec
„ Fletcher, Russell, & Co.
„ Waygood & Otis, London.

WILFRID J. L

Goldsmiths' Institute,
March, 1904.

PREFACE TO THE EIGHTH EDIT

THIS Edition has been considerably enlarge to still keep the book fully abreast of and the following firms and gentlemen have n assisted me with matter for Appendix VI.:—

Messrs. Samuelson & Co. Banbury, Pfeil & C Alfred Herbert, Ltd., Coventry, Herbert Morris and Crossley Bros. Messrs. Andrew Forster, C ball, J. Hartley Wicksteed, F. E. Kennard, W. W. F. Pullen; to all of whom and to othe tender my sincere thanks.

WILFRID J. L

Goldsmiths' College,
November, 1905.

PREFACE TO THE NINTH, TENTH, AND ELEVENTH EL

GENERAL revision and correction only required.

WILFRID J. L

Goldsmiths' College,
August, 1906, *September*, 1907, *August*, 1909.

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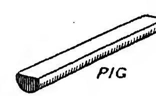
PART I.—WORKSHOP PRACTICE.

CHAPTER I.

CASTING AND MOULDING.

UP to the time of Watt, and even later, a very great deal of wood was used in engineering structures, even to the extent of steam pipes, but as fluid pressures became higher, other materials were sought, and cast iron was the first to recommend itself.

Cast Iron is the most crude form of the metal, and is obtained direct from the blast furnace by the fusing of the ore with some flux, which varies according to the nature of the particular ore, sometimes requiring clay, but in this country usually lime. The molten iron runs down into channels or *pigs* and is then called *pig iron*, while the slag is withdrawn separately.



Of the pig iron thus formed there are eight commercial varieties, according to the quality of the ore and the blast used; thus, increase of blast and diminution of fuel gives a whiter iron.

Commercial numbers.	
8, 7, 6, 5	**White** (silvery, hard, and strong), for conversion into wrought iron.
4	**Mottled.** Strong foundry iron.
3, 2, 1	**Grey** (soft and weak) for ornamental castings.

Most of the impurities disappear in the blast furnace, but carbon is absorbed from the coke fuel, and the presence of this carbon, *mechanically* mixed in the form of graphite, makes the iron more liquid when molten, but at the same time produces weakness in the casting. There is never more than five per cent. of uncombined carbon, while in the white iron there is almost none, it being *chemically* combined, and then actually increases the strength of the iron.

Table showing chemical composition of the three principal varieties of pig iron, in percentages:—

	Grey.	Mottled.	White.
Iron	90·24	89·3	89·86
Carbon (combined)	1·02	1·79	2·46
Graphite (uncombined)	2·64	1·11	·87
Silicon	3·06	2·17	1·12
Sulphur	1·14	1·48	2·52
Phosphorus	·93	1·17	·91
Manganese	·83	1·6	2·72
	99·86	98·62	100·46

The Cupola.—The pig iron is re-melted in the foundry in a kind of small blast furnace called a *Cupola*. The cupola is re-lit every day (and is therefore not so economical as a blast furnace, where the fire is never allowed to die out),* but this cannot be avoided on account of the intermittent demand made upon it.

Fig. 1 is such a cupola, where the pigs and coke are raised by the lift H L, hydraulic or otherwise, together with the man, who, after breaking each pig in three, puts them all in at the door D, charging as follows:—First, 7 cwts. of coke, next 1 ton of iron; then, alternately, 2 cwts. of coke and 1 ton of iron, until the cupola is filled to D. The blast enters at B, and the mouth M is stopped with luting clay. When all the iron is melted M is tapped, and the metal taken away in ladles to the moulds.

During re-melting the iron is again apt to absorb impurities from the fuel, such as oxides and silicates, the latter especially producing more brittle material, and rendering the iron cold-short, that is, easily snapped when cold. Formerly, re-melting was believed to be an improvement, and founders were advised to

* One blast furnace in the North of England, known to the writer, burned for over twelve years incessantly, and was then only blown out for repairs.

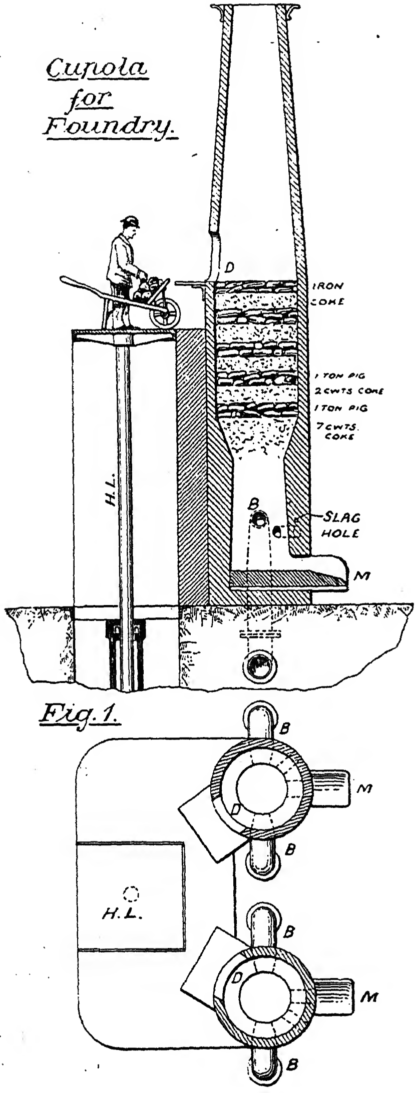


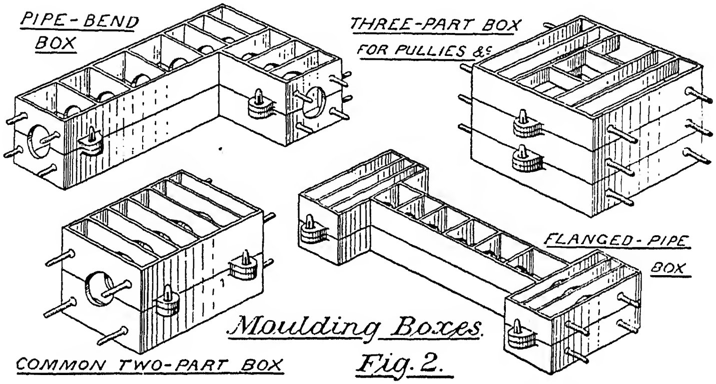
Fig. 1.

melt again and again, even to twelve or thirteen times, but this has now been demonstrated to be a fallacy (*See p.* 779 *and* 1011.)

To obtain a very tough casting, such as an hydraulic cylinder, wrought iron turnings are sometimes mixed with the pig in the cupola.

Moulding.—We will now consider how the moulder forms his casting into any desired shape. To do this it is necessary in most cases to make a wooden pattern which shall be the counterpart of the casting required; for several reasons we shall see, however, that the pattern will not always be exactly similar to the casting. But more of this as we advance.

The pattern is impressed in sand contained in two moulding boxes, or flasks, about half the pattern in one box, and half in the other;



Moulding Boxes.
Fig. 2.

these are sketched at Fig. 2. The boxes are light castings, ribbed across as shewn, allowing space for the escape of gas from the molten metal. (*See Appendix II., p.* 781.)

Sand used in moulding is of two kinds, green sand and loam. **Green sand** is obtained from the chalk or coal measures, that of the London basin being among the best. Green sand should contain a large percentage of silica to give porosity, together with a very little magnesia and alumina for binding purposes. The lining of Bessemer converters has about 85 per cent. of silica in its composition, while many moulders prefer to have as much as 93 or 96 per cent. of silica, leaving only 4 or 7 per cent. of other substances. The sand should not burn on setting, or it will stick too much when wetted for use again, and, while cohesion is necessary, it should at the same time be porous enough to allow for the passage of air, though not so much as to permit of any molten metal entering it.

Loam is a mixture of clay (ferruginous or calcareous*) with a considerable amount of rock sand (abraded rock). It is ground in a mortar-mill and mixed with powdered charcoal, horse dung, cow hair, chaff, &c., to give it binding power and porosity.

Besides the above, **Cores** require a mixture of rock sand and sea sand (the latter for porosity), and **Parting Sand**, for the use implied by its name, consists of finely powdered blast-furnace cinder, brickdust, or fine dust from castings; all perfectly dry.

Moulding is practised by three different methods: *Green Sand*, *Dry Sand*, and *Loam Moulding*.

* With iron or lime respectively.

In green and dry sand moulding, patterns are generally used; but in loam moulding, which is only employed for objects of regular form, the mould is struck out by means of a template, and built up by the moulder himself.

Green Sand is the geological name of a sand of very fine texture. It appears black in the foundry because it is mixed with a proportion of coal and charcoal dust; it is damped each time that it is used. This is the most general method of moulding, with castings not likely to warp too much by the more rapid cooling. (*See Appendix II., p.* 779.)

Dry Sand is a mixture of old loam with an addition of rock sand. It is so called because, after the pattern is moulded, the sand is dried by means of fires hung in pans or trays over the moulds. It is firmer and more suitable for the support of long castings, such as pipes, columns, and large fly-wheels than green sand is, and will produce finer castings, with less fear of pieces of sand being torn away by the flow of the metal. If pipes are moulded in green sand, the tendency is to uneven thickness in the castings, through sagging of the sand.

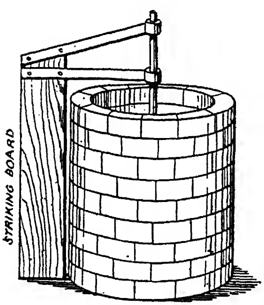
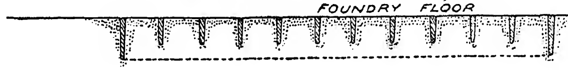


Fig. 3.
Loam work.

Loam Moulding, as we have said, does not require a pattern, the mould being struck in the pasty loam (the latter being mixed with water) by means of a rotating or sliding template, called a striking-board. Thus the core of a large cylinder is built up in brickwork, and then covered with a layer of loam, which is smoothed by a rotating striking-board (see Fig. 3), much as a plasterer would work the cornice of a house ceiling. Cubical moulds, such as those for condensers, may also be worked in loam.

The simplest moulding done in *green sand* is called **Open Sand Moulding,** and consists in laying the pattern in the sand on the foundry floor, withdrawing, and then pouring in the metal, a cover not being used. This is the method employed for such common objects as moulding boxes (see Fig. 4).

Fig. 4.



Opensand Moulding.

A Cattle Trough.—Our next example of mou
ordinary cattle trough, and here two boxes are used
sand. The pattern may be of wood in the first inst
the same shape as the finished casting. It is pl
bottom box, as in Figs. 5 and 6, and sand is filled in to
which is the parting. This parting is smoothed off,
parting sand applied, and, the top box being put on
down, the whole is filled with sand and rammed w
The top box must now be removed and the pattern
a slight rapping being given to effect its detachme
√ sand, while the latter is dusted with **Blackening,** w
charcoal dust. But first, to make the blackening ad
meal is sprinkled on the mould, absorbing the damp
and thus becoming a pasty layer. The object of the
is this: If the metal were to touch the side of the mo
enter into the sand surface, and thus produce a ro
This is allowable in moulding boxes, where roughness
advantage, but where a smooth casting is desired, b
needed, as it ignites on being touched by the metal, a
a film of gas between it and the mould, a clean castir
result. (*See pp.* 747 *and* 1012.)

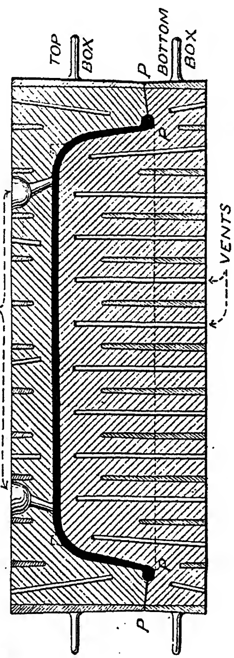
Gates.—The mould having been sleeked and f
any little break in the sand mended, the gates hav
made for the entrance of the metal. Tapering plugs
usually left in the sand for that purpose, and these are n

The more shallow the casting, the more gates a
many even as four.

Vent Holes are made in the more solid parts
(but not to touch the surface of the mould) to f
passage of air from the latter.

The moulders, being provided with molten iron,

the cupola in ladles, as already described, pour it simultaneously into the gates of the mould, and the sand being afterwards broken away, reveals the casting, which has filled the matrix left by the pattern.



Cattle Trough.

Fig. 6.

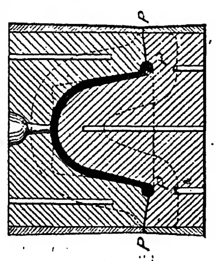
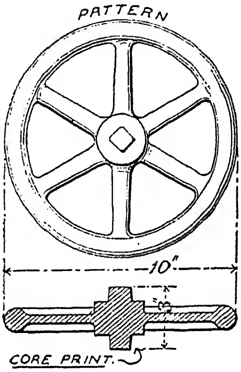


Fig. 5.

As regards the proper position in which to lay the pattern, a little thought is necessary, but as a general rule the most unimportant part of the casting should be upward, that being the part to which the scum and impurities rise. If possible, the scum should be entirely removed from the mould itself, being allowed to fill a large gate or projection. This is especially done in the case of steam cylinders, where purity is a necessity. Gates should be as central as possible, and have their mouths a little *higher* than the mould, but they should, as a rule, enter the latter low down, particularly in deep castings, in order that the air may be made to pass out at the vent holes; but much judgment has to be exercised; and in most cases they should be placed a little on one side, namely, not to enter on the top surface, otherwise the corners of the sand may be knocked off by the force of the flow; and finally, they should be put where shrinkage is likely to occur, that they may tend to fill up any shrinking portion.

Our next example shall be a **Hand Wheel** for a large stop



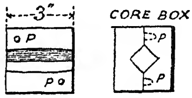


Fig. 8.

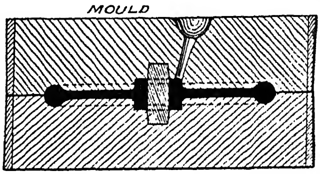


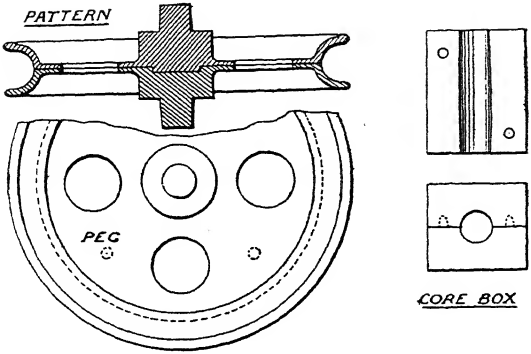
Fig. 7. Fig. 9.

Hand Wheel.

valve. The pattern here will be of the same shape as the finished casting, excepting the square holes in the centre, which we will suppose to be cast in, to be afterwards dressed up with a rough file. Square core prints of any convenient length are put on either side of the boxes, and of a diameter equal to the hole to be cast, allowing a small amount for cleaning up (see pattern, Fig. 7).

A core box is now to be made, which is shown in Fig. 8, and consists of two blocks of wood, hollowed out in such a way as to represent the square hole required, and of a length equal to that across the core prints from end to end. The pattern is next placed in the sand, as drawn in Fig. 9, and parting made, then rammed up, with gate at G, and vents here and there. Core sand mixed with water is put in the core box (which fits together by means of the pegs PP), smoothed off, removed, and placed in the core stove to dry. The mould is finished and blackened, and the core treated with *black wash*, which is charcoal dust mixed with clay water, and used for the same purpose as blackening. The core being put in place, as shown in Fig. 9, everything is ready for the reception of the metal.

We shall now take the moulding of a **Chain Pulley**, by a very ingenious method. Fig. 10 represents the pattern, with core



Chain Pulley. Fig. 10.

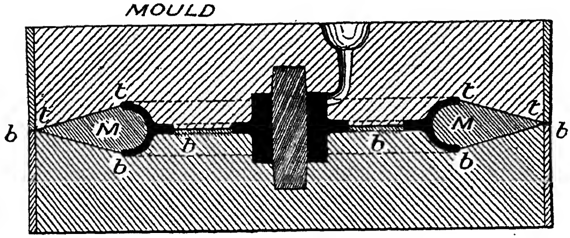


Fig. 11.

prints for the centre hole, and divided in halves by a horizontal plane. In all other respects it is the counterpart of the casting. The operation is as follows: referring to Fig. 11, we must first lay the bottom half of the pattern in the bottom box, and make the parting *b b*; next put in the top half of pattern, and make the parting *t t*; lastly, fill up the box, and ram well together. The pattern has now to be drawn out, and this is done by first lifting off top box, taking away the top half of pattern, and returning top box. Now, on turning the whole upside down, the bottom half of pattern becomes the top, and may be similarly released. The ring of sand M, it will be noticed, has all this time remained resting on that half which happened to be at the bottom. It is only necessary to make the core as in last example, put it in

place by removing top box, form gates and vents, and complete casting.

A Worm Wheel may have a pattern made in halves, and be moulded in an exactly similar manner (see Fig. 12) ; the teeth on

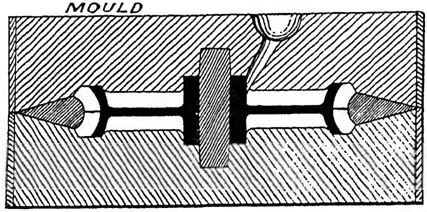


Fig. 12.
Worm Wheel.

the pattern being formed so as to gear with a wrought iron worm which has been previously turned, the worm and wheel pattern being rigged up on two axles to imitate their condition when in actual work. In withdrawing the pattern from the sand a slight screw motion must be given to allow for the angle of the teeth.

Moulding boxes are entirely or to some extent dispensed with, and the floor of the foundry used for the reception of the pattern wherever convenient; and then, except in such cases as that shewn in Fig. 4, a cope or slab of sand is used, contained in a box, to cover the impression. Examples of this kind of moulding, with more or less complicated copes, will now be treated.

Fig. 13 is the plan of a **Drilling Machine Table.** The pattern is of the same shape, with the exception of core prints necessary for the slot holes. A core box is required for these holes, and the whole is moulded face down, an extra piece being left in the casting at top, if thought necessary, to allow for scum. Instead of the bottom box the floor might have been used, if previously well vented with coke.

Perhaps the most ready way to mould the **Cylinder Cover** shewn in Figs. 15 and 16 is to use three boxes (or what really comes

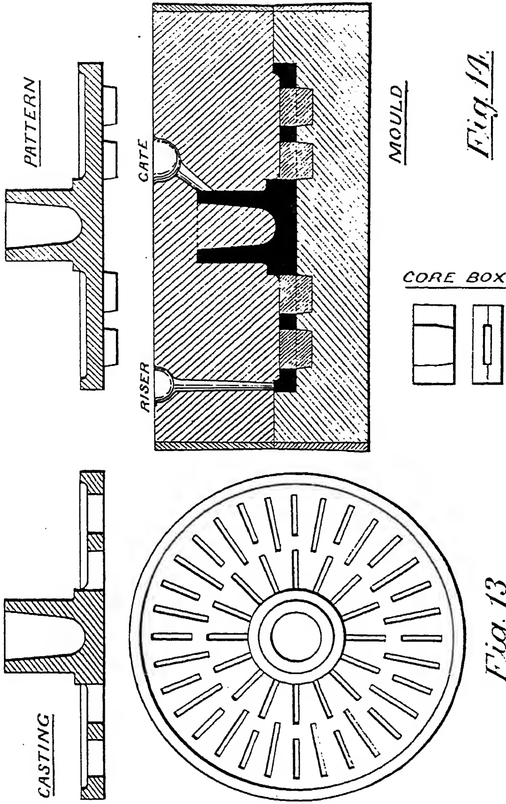
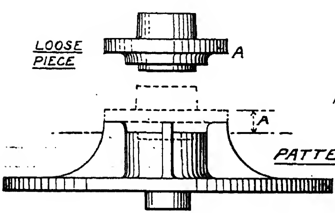


Fig. 13

Fig. 14.

Drilling Machine Table.



Cover for Steam Cylinder.

Fig. 15.

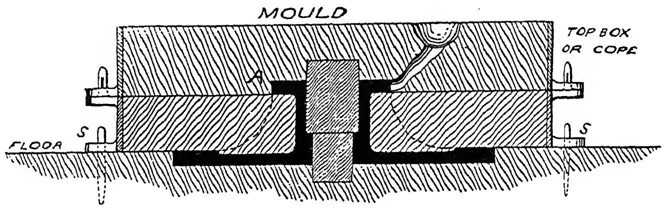


Fig. 16.

to the same thing, two boxes and the foundry floor), and make the flange A loose on the pattern. On taking off the top box, this flange may be withdrawn, while on replacing the top box, and lifting it and the bottom box together, the main pattern may then be removed. The core for the centre is inserted in the usual way, and the casting made in the position shown. The stakes S S fix the position of the boxes with regard to the floor. (*See App. II., p.* 781.)

Casting on.—Sometimes it is necessary to attach cast iron to wrought iron in the mould itself, and so do away with the expense of bolts. *Casting on* is the term adopted for the operation resorted to.

As an example, we will take the traction engine **Road Wheel**, shown in section, Fig. 17.

A core box is made as in Fig. 18, consisting of a slab of wood A, with the boss B fastened to it, and of a hollow cylinder of wood C to contain the core sand. Two cores are thus formed and baked in the stove. A second core box is required, shown in Fig. 19, consisting, as before, of a hollow box to hold the core, and of the bosses D D in two parts, to make the impression for the central part of the wheel nave. As the line x in Fig. 18 corresponds to line x on Fig. 19, it will be seen that the prints P P in Fig. 18 will leave spaces for the reception of the spokes.

It is only necessary to fix the spokes loosely in place by bolts to wheel-rim, at the same time laying them in the spaces left for them in the cores; build up according to Fig. 20; add a central core, E, made in ordinary core box; make gates, and cast. The spokes are afterwards riveted on wheel rim, and have the shape shewn in Fig. 21.

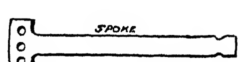


Fig. 21.

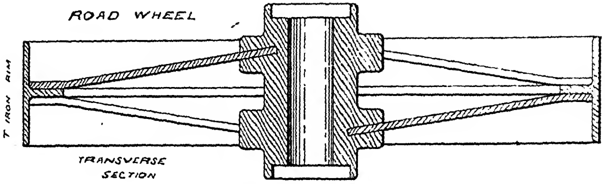
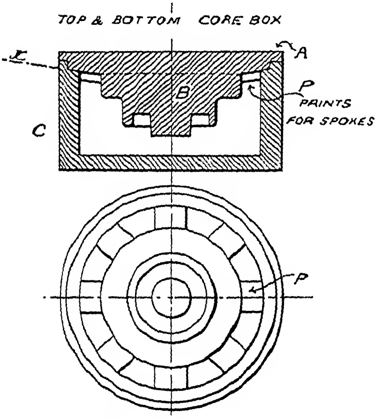


Fig. 17.



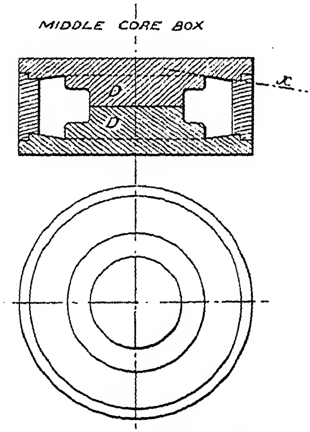


Fig. 18.

Fig. 19.

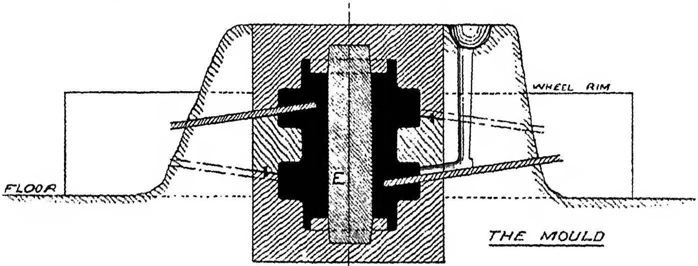


Fig. 20.

Road Wheel for Traction Engine.

Loam Moulding.—We shall now proceed to consider the moulding of such objects as may be done wholly or partly in loam by striking (or strickling), and first we will take an ordinary **Gas Pipe Main,** with spigot and faucet, the former being the smaller end of the pipe, which fits loosely into the faucet or larger end of the next pipe, see Fig. 22, which represents the finished pipe in section. To mould the outer envelope, we may either

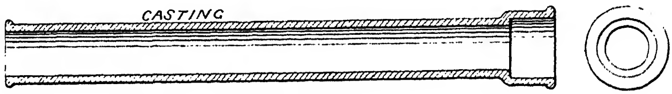


Fig. 22.
Gas-pipe main.

have a wooden pattern with the core prints at the ends, or may strike out a loam pattern from a board. Assuming the latter method, we need first a pair of trestles, Fig. 23, on which is placed a hollow cast iron cylinder with journals at the ends, and pierced with holes along its length for the venting of the core.

Round this cylinder straw rope is tightly coiled, after which a layer of loam is laid on. The loam being dry, a second coating is applied, and this time, as the handle is turned, the shape of the core is struck out by means of a board B secured to the trestles. The core *b* being dried in the stove, is blackwashed, and then covered with another layer of loam to be struck out by the second board A, and so the loam pattern is formed. Being again dried, an impression is made in the mould, after which, the last applied loam, or *thickness piece*, is removed, the blackwash facilitating this, and the internal core *b* being returned to its proper place in the mould (see Fig. 24), gate is made, and casting performed as usual.

It is advisable to cast these pipes either vertically, or on an incline, so that the metal may flow more easily and bring the scum to the end, and if they are very long, dry sand should be used in the boxes instead of green sand, for reasons previously stated. After the metal is poured, the escaping gas is lit at either end of the pipe. (*See Appendix II., p.* 782.)

Fig. 25 represents the moulding of a **Bend** for the pipe in last

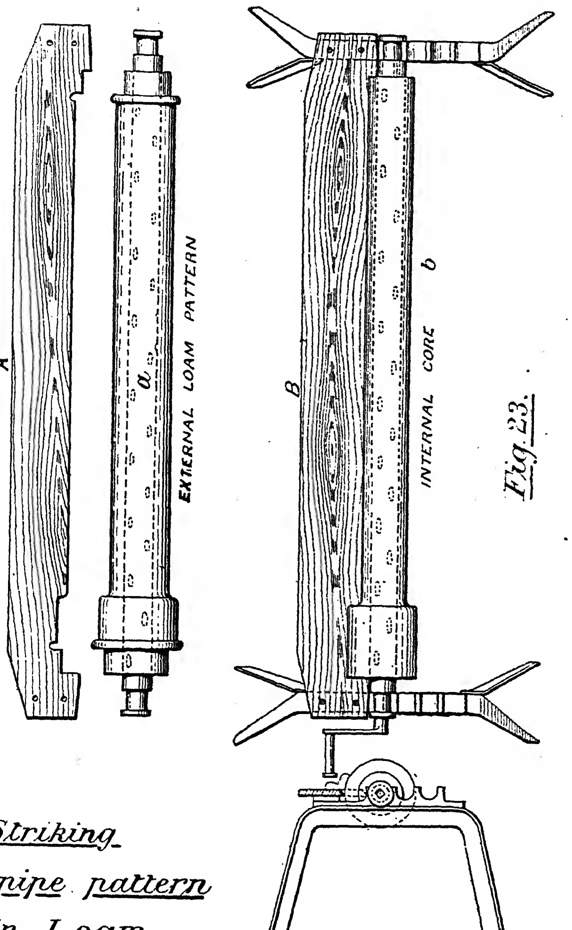
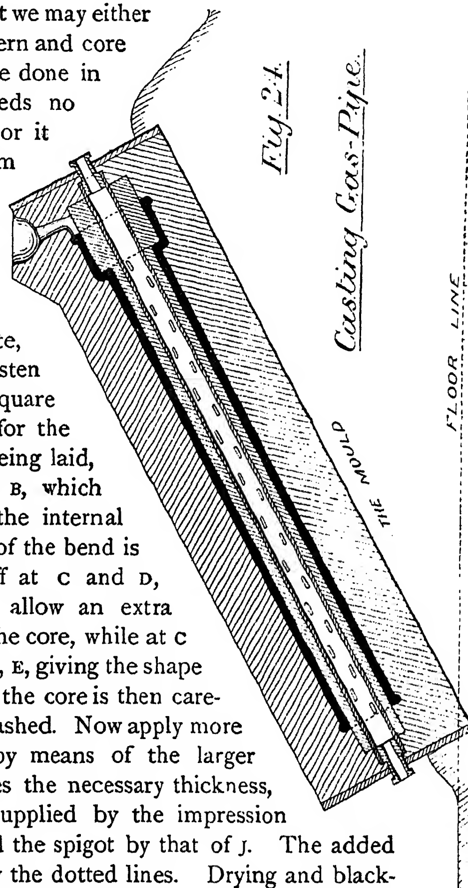


Fig. 23.

Striking Gas-pipe pattern in Loam

example. To mould it we may either have a complete pattern and core box, when it would be done in green sand, and needs no further explanation; or it may be worked in loam by the aid of templates. For the latter method we may proceed in the following manner:



Take an iron plate, Fig. 25, and on it fasten the bent wire A of square section as a guide for the template B. Loam being laid, it is traced out by B, which gives it the form of the internal surface. The length of the bend is carefully measured off at C and D, care being taken to allow an extra piece at D to support the core, while at C is applied the core box, E, giving the shape of the internal faucet; the core is then carefully dried and blackwashed. Now apply more loam and trace out by means of the larger template G, which gives the necessary thickness, while the faucet is supplied by the impression of the core box H, and the spigot by that of J. The added thickness is shown by the dotted lines. Drying and blackwashing are again performed. Lastly, cover the whole plate with loam, which, as it is required to be lifted off entire, must be well stayed with cross and longitudinal wires tied together by small wire. A wire should also stiffen the internal cores. (The whole of the foregoing is repeated opposite hand, on the plate F, the same bent wire being used.)

The solid block of loam thus formed on either board is taken off, dried and blackwashed, and now all is ready for putting

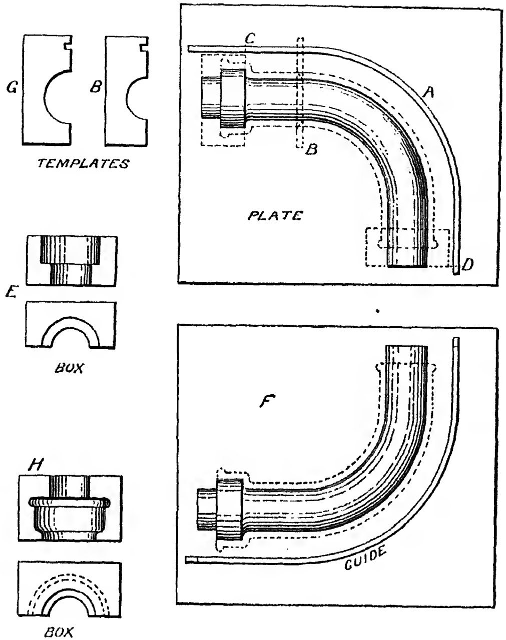


Fig. 25. Gas-pipe Bend.

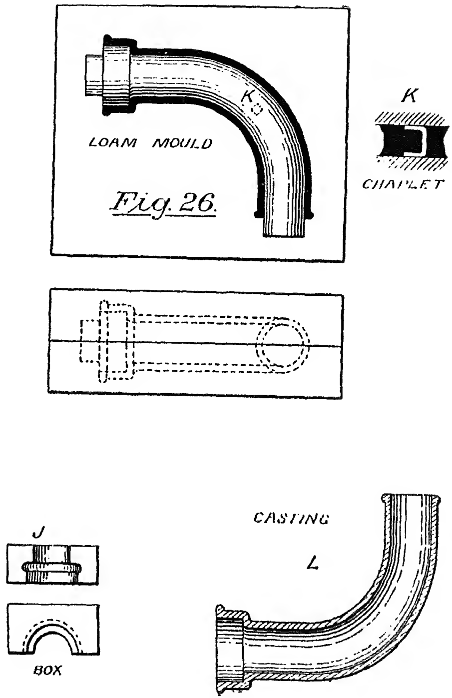


Fig. 26.

together to form the mould, with the exception of the tl
layer on the internal core, which must first be removed, :
two half cores taken off the plate and put back to bac
forming a complete central core. The whole mould is
at Fig. 26, the bent part of the core being supported by a
given in detail at K; being a bent, tinned, plate.

It must be quite understood that if many castings are re
the above operation would not be performed, as a patterı
give more expeditious results.

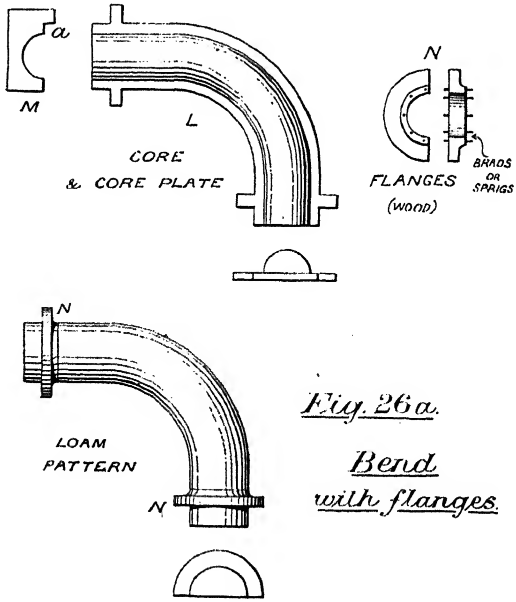


Fig. 26 a.

Bend with flanges.

A slightly different, but more usual way of moulding
bend, is to cast two plates, as at L, Fig. 26A, from a
pattern of the shape of the pipe, a little margin being left
side, the figure being for a pipe having flanges at the end;
template takes the form M, and has a few nails driven in

prevent considerable wear as it travels along the plate. The internal core is shown struck on the plate, and the opposite hand having been made, these two are dried and laid aside. A larger

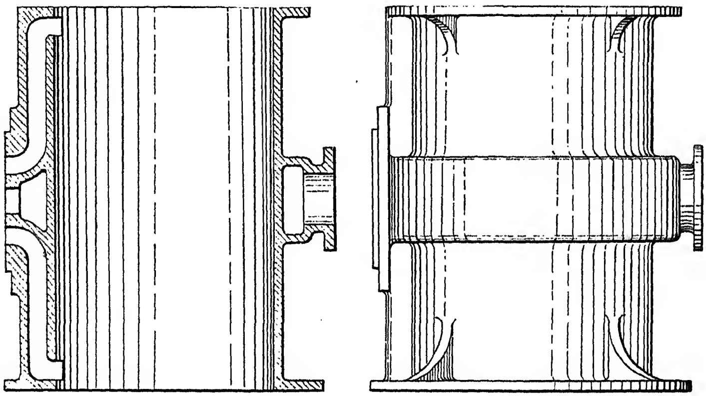
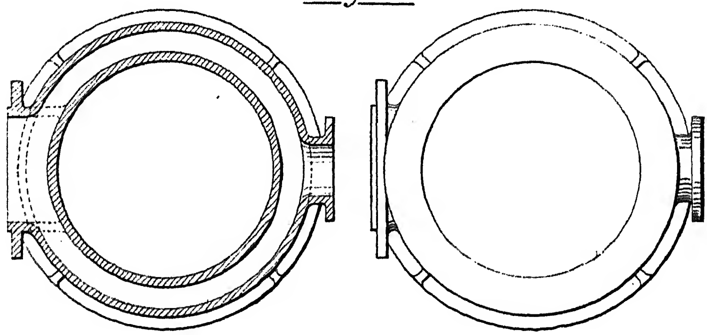


Fig. 27.



Steam Cylinder.

THE CASTING.

template strikes out the patterns, right and left hand (the wooden discs N serving as flanges), and these being dried, it is clear that we now have two half loam patterns and two half cores; the moulding, therefore, may take place in green sand in the usual

Steam Cylinder.

STRIKING THE MOULD

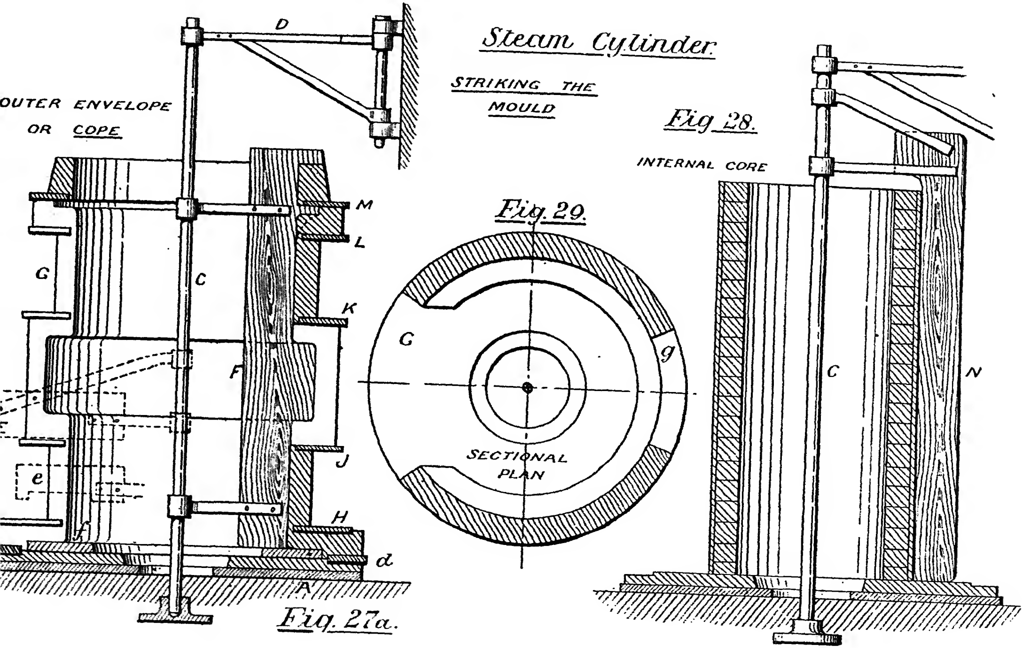


Fig. 27a.

Fig. 29.

Fig. 28.

way. A few sprigs or brads in the flanges N serve to fasten the latter to the pattern.

The mould for a **Large Steam Cylinder** is usually made entirely in loam, and this operation we will now examine.

Fig. 27 represents the casting in longitudinal section, elevation, and in plan. The valve box is made separately, as is sometimes done with these large cylinders, but in any case, no further explanation is needed beyond that previously given, as a pattern would be made for it. The body of the cylinder would be swept out entirely by template boards, but special projections, such as steam ports and exhaust flange, would require core boxes and patterns.

An iron plate A, Fig 27*a*, is laid on the foundry floor to support the structure, and a centre B is sunk beneath the ground-line, an upright spindle C being taken of sufficient length, and supported at the top by means of an arm D standing out either from the wall or from a crane pillar; all is now ready to begin.

A base of loam is swept out by the board E, shown in dotted lines, and representing the bottom of the cylinder flange; this is dried and blackwashed, a flat ring *d* being then laid as a foundation for the core structure.

Taking board E away, another (*e*) is used to strike out the lower cylinder flange *f*, which is necessary as a support to help plate *d*.

The loam *f* being dried and blackwashed, the external core of the cylinder is next formed, because it is necessary to remove it for the formation of the internal core, and the latter, being in one piece and cumbrous, is made separately. The board F is now used to strike the outer form, the central projection being for the exhaust port, and an opening must be allowed at G, the full length of the cylinder (see Fig. 29) for the reception of the port cores on one side, and which may be traced out by a template board, while a similar opening *g*, of the depth K J, must be left on the opposite side for the exhaust flange core. It will be noticed that this outer mould requires, for building, the aid of annular plates at H J K L M, for the support of different pieces of the structure. These plates do not go entirely round, being prevented by the ports at G, and

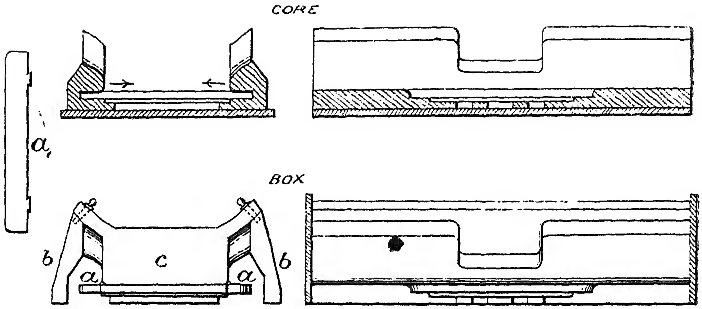


Fig. 30.

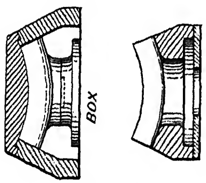


Fig. 33.

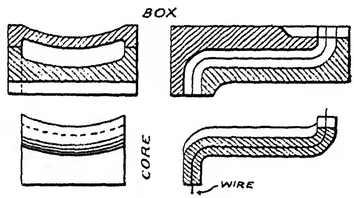


Fig. 31.

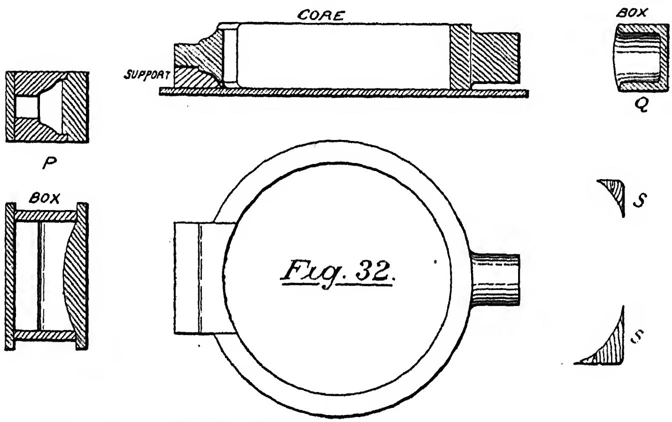


Fig. 32.

Steam Cylinder.

CORE BOXES

they now enable us to remove this outer portion in separate pieces to a safe place to dry, and allow us also to build up the internal core. Thus plate K may be lifted by crane, removing the upper portion first; next J and *d*. Discarding the loam plate *f*, which is no longer needed, our next proceeding is to take the board N, Fig. 28, having built up the core loosely with bricks, vented at the joints with coke powder, and strike out loam to represent the internal surface of the cylinder; this is dried in place by open fires and blackwashed. The projecting portions only now remain, which, as we have said, must be made from core boxes. Fig. 30 represents the box for the outer contour of the steam ports, and a core is formed by laying it on a flat plate and filling up with loam. The parts *a a*, of the core box should be noticed: sides *b b*, can be easily taken away, but in order to draw away the centre *c*, the flanges must be dovetailed to *c* in such a manner that they may be left behind on withdrawal of the box.

This may be understood on reference to *a*, which shews one of these *loose pieces*. They may afterwards be taken away in the direction of the arrows. The box and core for the steam ports are shown in Fig. 31, and need no explanation.

The inside core for exhaust port, being circular, is struck out on a separate plate by board (Fig. 32), box P being required to give the projection on the steam side, and Q for that on the exhaust side. There is left the exhaust flange, which may be formed from the box in Fig. 33, the flange itself being loose on the pattern to enable the core to be withdrawn, the latter being made on a plate similarly to Fig. 30.

S S are patterns for the web at top and bottom of the cylinder, which, having been built into the core at Fig. 27*a*, may now be removed.

Finally, all may be put together to form the mould, in the manner drawn in Fig. 34, beginning at the bottom and putting the different cores in their places as we proceed; chaplets being required to support the annular exhaust core. Gates are next made, which had better enter the mould somewhat low down, in order to have some head of metal at that point. The object of this is to prevent air bubbles in the casting, by means of the weight of in-pouring metal, whatever air there may be in the

mould being thus forced upward, where it escapes at holes there provided, termed *risers*. The mouth of the pouring gate should of course be a little higher than the top of the cylinder. Venting is rarely necessary in loam moulding, except for such pieces as the long **S** cores. Piece R, Fig. 34, is left for the purpose of receiving

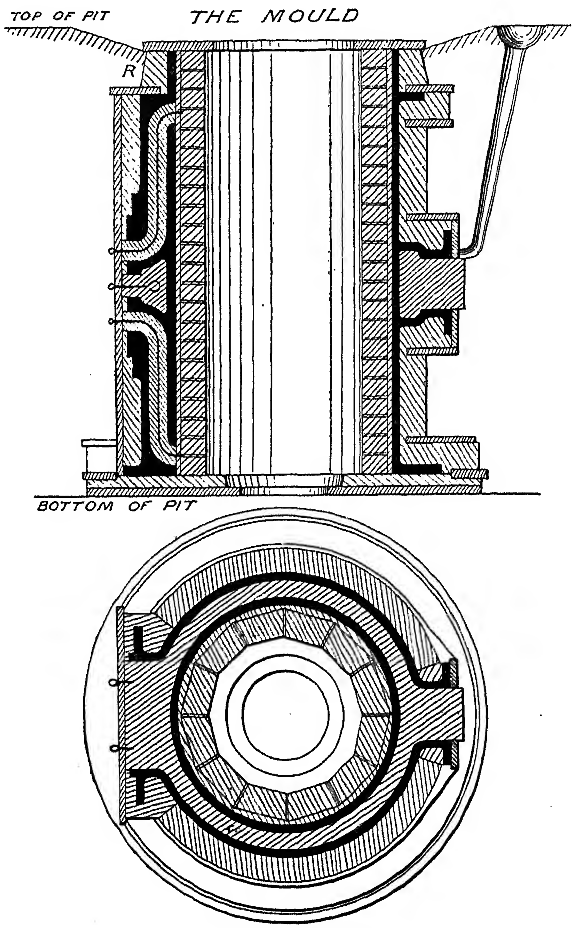


Fig. 34. Steam Cylinder.

the scum, in order to leave the casting sound. Wherever the molten metal is to touch the iron plates the latter should be washed with loam.

Foundry Pits.—It should be understood that the floor we have spoken of in the last example is not strictly the foundry floor, but that of a pit deep enough to hold the whole mould. Important castings like the one last considered, and especially upright ones, are always thus treated, and after the mould has been finished the space left in the pit is filled in and rammed so as to bed the mould tightly against the sides of the pit, and thus resist the pressure of the metal on casting.

A Screw Propeller can be best moulded in loam, a pattern being provided for the centre boss. Referring to Fig. 35, a board A centred on the vertical spindle, and balanced by means of a small weight, is revolved so as to travel along the incline B C, which is only a template curved so as to have D as its centre, and forming part of a screw of the same pitch as the propeller. It is very clear then, that by backing up the surface B C with loam, we shall obtain a screw surface the same as that of the propeller blade required. The next thing is to mark out the shape of the blade, shewn in dotted lines. On the blade thus marked out, dried and blackwashed, we now lay strips of wood, as shewn at *e*, Fig. 36, representing the thickness of the propeller blade, and the surface is then covered with loam up to this thickness, smoothed off, and again dried and blackwashed. Now completely cover with loam, and so form top mould, which in its turn is taken away and dried. The thickness piece being removed, the blade is completely moulded, and this may be repeated for the other blades. Setting all the lower moulds then in position on the floor, the bottom half of boss pattern is applied (Fig. 37), and, being filled round with dry sand at E E, the top half is treated similarly. Lastly, the mould is completed by the addition of a core for the central hole, and of the top box, and the whole has the appearance of Fig. 38.

A large **Fly-wheel** may be moulded without the necessity of making a pattern for the whole of it. A coke bed is first formed on the floor for the purpose of venting, and a centre is sunk for the spindle A, Fig. 39. Then the core box in Fig. 40 is taken, which is formed so that a certain number of cores made

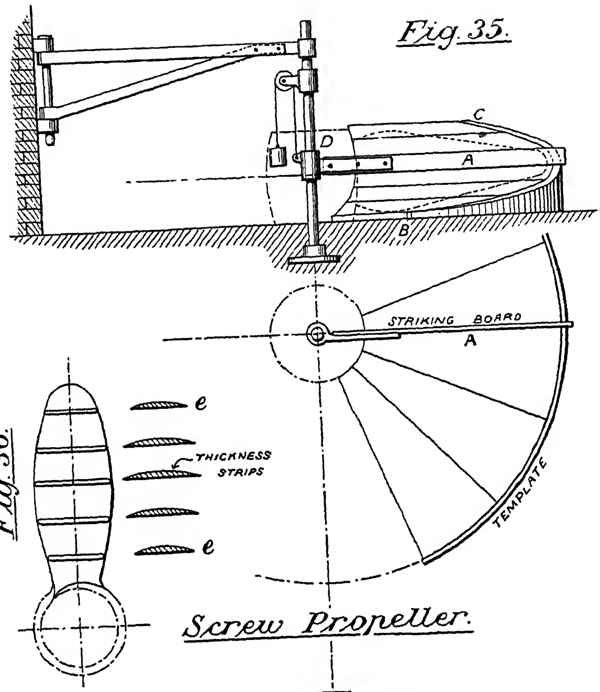


Fig. 35.

Fig. 36.

Screw Propeller.

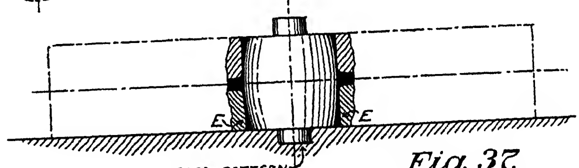


Fig. 37.

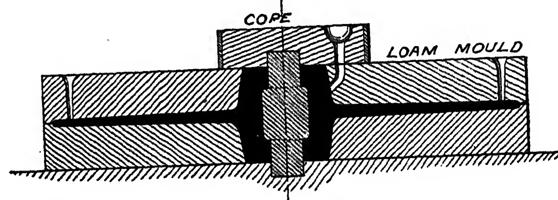


Fig. 38.

from it may reach quite round the outer rim of the wheel, as suggested by the dotted lines; the back and top boards B and C being loose, to remove the core, which may be made in dry sand. After levelling the floor by means of board D, Fig. 39, the cores from Fig. 40 are set up at E, with the curved surface inward and gauged from the centre by the striking board F, which has the same radius as the outside of the fly-wheel rim. A small space is, however, left for the application of a coating of loam which is struck out at top and side by the board F.

We next require the arm cores. The box for these is shewn at Fig. 41, and supposing we have in our case six arms to the wheel this box must be made a sector of one-sixth part of the circle. The top, bottom, and sides are removable, so that when the box has been filled with compact 'dry sand' they may be taken away, together with the rim part and boss, leaving the arm, which, being tapered, may be knocked out with a mallet at G and so removed. Some moulders might prefer loam for these cores, which would be baked in the usual way.

Putting the sector cores in place, as in Fig. 42, a pattern is used for the boss, with a cope of green sand at H. There only remains the completion of the cope for the rim. This may be done in dry sand, contained in boxes shown in plan at J, and by means of a pattern K placed in the channel formed for the rim, top box J being put on and rammed up there. This pattern K is passed round until the whole of the top of the rim is formed, and is finally withdrawn by removing one of the boxes. The mould is now complete, and it is only necessary to form the gates, which should be pretty central, while risers (about four) are put in the boxes J to shew when the metal has filled the rim, which is known by its lifting a metal ball placed upon them. Of course great care must be taken in finishing the mould, so that no unsightly marks be left on the casting at places where the cores join each other.

Marine Condensers, being usually large cubical castings, are built up in loam in the manner described for other objects, by the aid of what are known as skeleton patterns, projecting flanges having patterns and core boxes. (*See Appendix II., p.* 782.)

Fig. 43 shews one or two objects suitable for loam moulding, A being a large **Air Vessel** for a pump, and B a **Cone Pulley** for some machine tool.

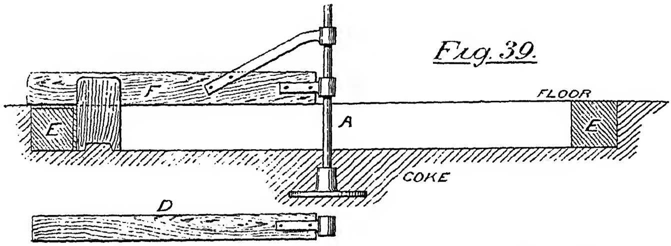


Fig. 39.

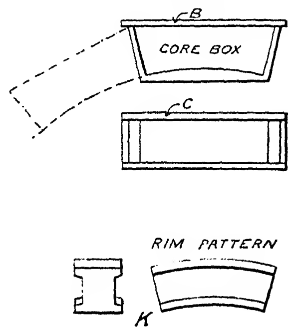


Fig. 40.

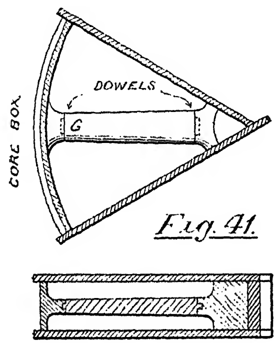


Fig. 41.

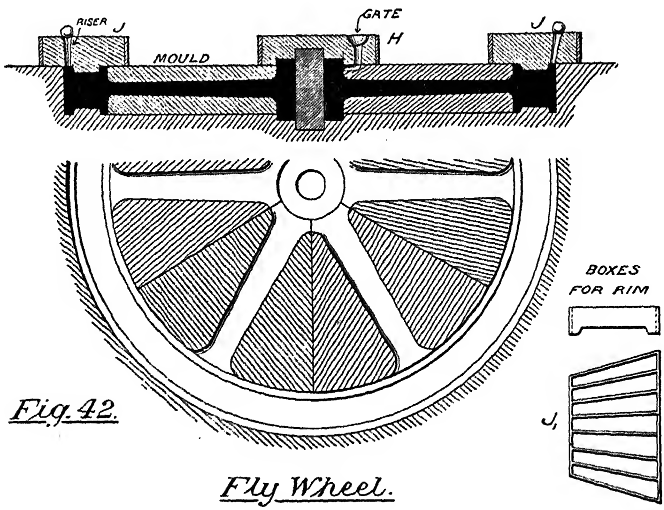


Fig. 42.

Fly Wheel.

The internal cores may be noticed in these examples. They are in both cases supported by being hung from the top-plate. In the air-vessel A, the portion *a* is struck out first, by means of

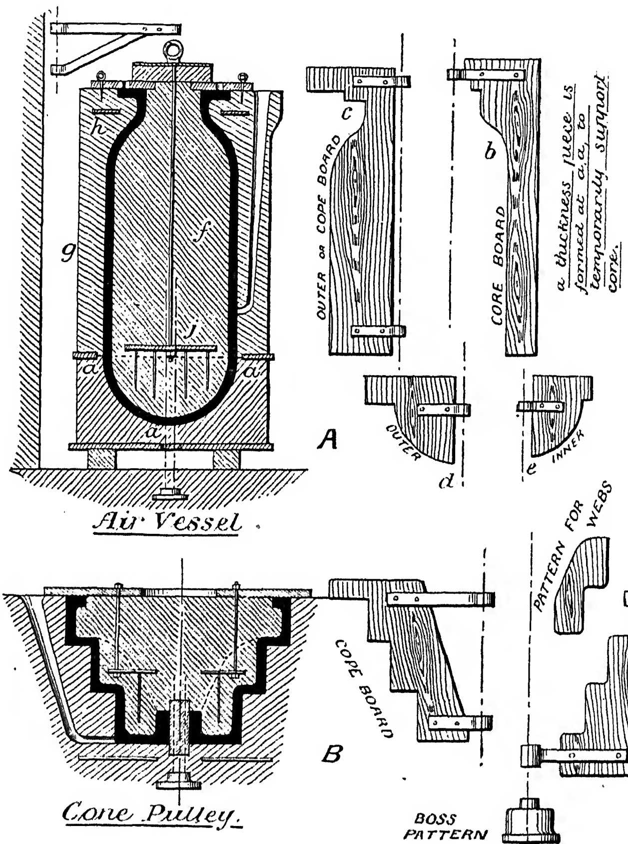


Fig. 43.

boards *d* and *e*, with a thickness piece, blackwashed as usual; then follow with internal core *f*, using board *b*. Remove *f* when dry, and strike out *g* by means of board *c*.

The core *g* is now removed, but returned in company with *f*, first taking away the thickness *a a*, and the whole structure is then

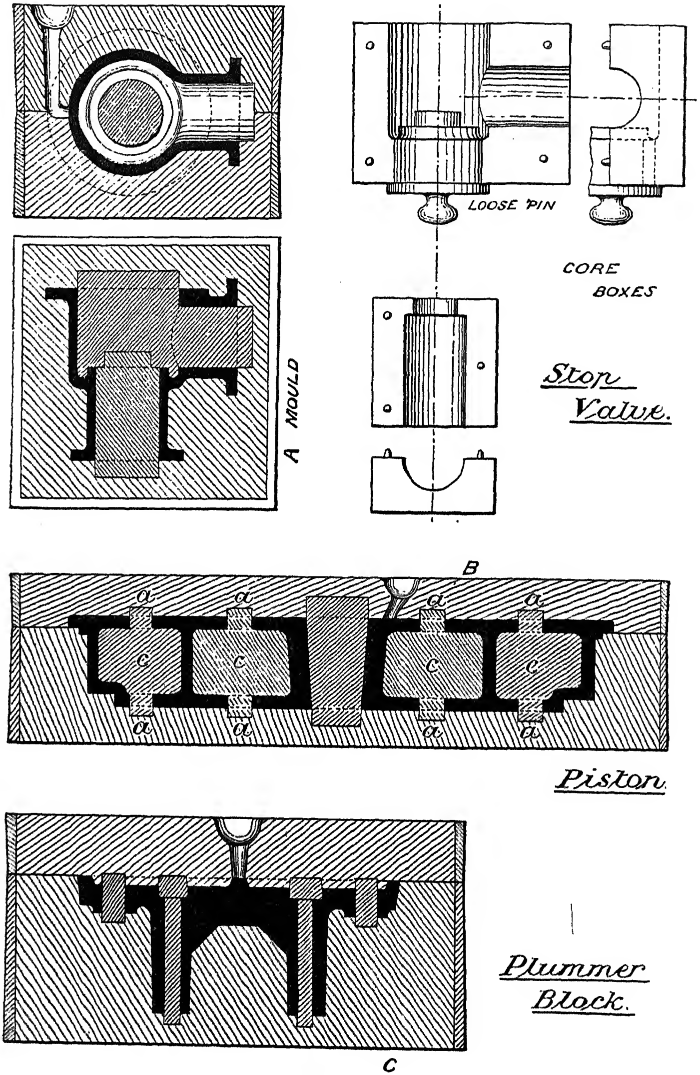

LOOSE PIN
CORE BOXES
A MOULD
Stop Valve.
B
a
c
Piston.
Plummer Block.
C

Fig. 44.

bolted together; *h* is a ring to stiffen *g*, and *j* is a pronged plate to support the core *f*.

The cone pulley B may be struck out in the floor, while the internal core is made separately on a plate; patterns being used for webs and boss. In fact, any casting of regular shape, either circular or cubical, may be moulded in loam much more economically than by means of wooden patterns; the symmetrical parts being struck, and projecting pieces having core boxes.

A few other examples of moulds in green sand for different objects, requiring no special description after what has been previously said, are given in Fig. 44, where A is a stop-valve, the larger core box for which has a loose pin to form the impression needed to support the smaller core: B a large marine- or stationary-engine piston, the core being supported by and vented through the pieces *a a*, which are filled in on finished casting by screwed plugs. Boxes are needed for the cores *c c*, and C represents the mould for a plummer block. (*See Appendix II.*, *p.* 784.)

Wheel Moulding.—Not many years ago all spur and bevel wheels were moulded by providing a finished pattern for the wheel required, but as machine moulding is not only simpler, but far more accurate, and as it does away with the necessity for storing heavy patterns, which are sure to be out of truth by the next time they are required, toothed wheels are now extensively moulded by machine.

Scott's wheel moulding machine may be understood by reference to Fig. 46, and it may be premised that three operations are necessary in the working of it.

A board B is set upon the central spindle A (see Fig. 45), for the purpose of striking out the greatest diameter of wheel, on which the teeth are to be formed, giving at the same time the height of the top and the bottom of the rim. The spindle being removed, the machine is put in the central socket C, Fig. 46, and the operations are now to be explained.

A pattern D, of two teeth, is accurately made in hardwood, and being fastened to the upright arm E of the machine, this arm needs to be—(1), fixed to the requisite radius of wheel; (2), raised or lowered; (3), passed round the rim of the wheel by the rotation of the arm F.

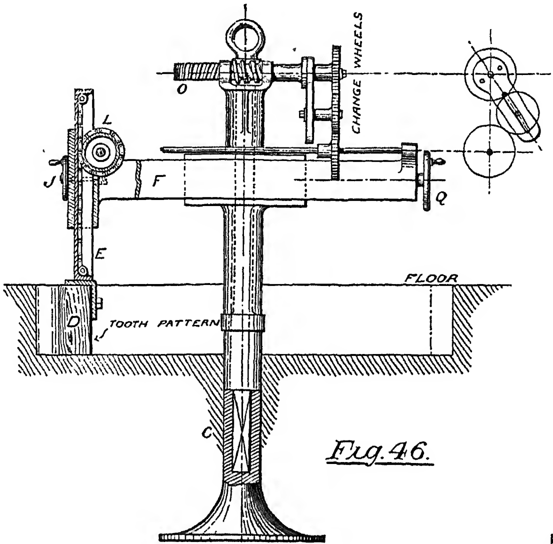
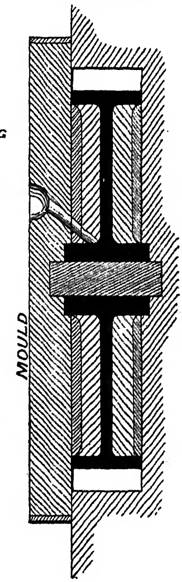


Fig. 46.

Fig. 47.



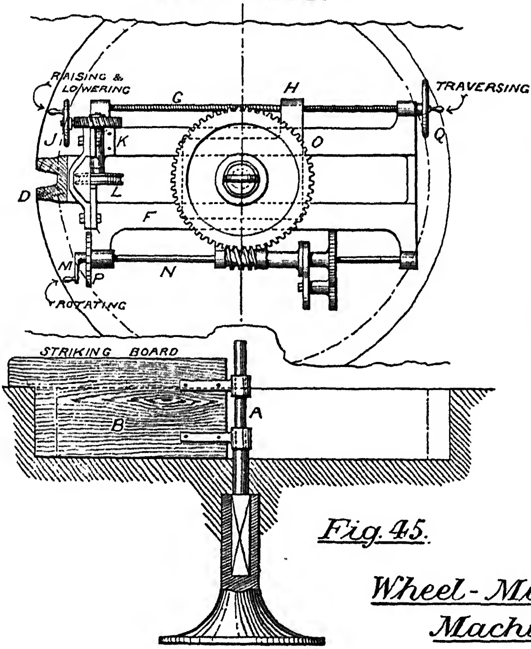
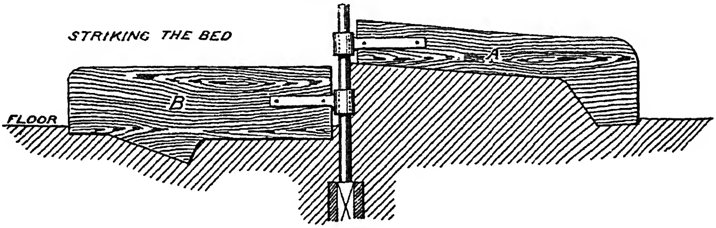


Fig. 45.

Wheel-Moulding Machine.

The first operation is done by means of the traversing screw G which slides the whole of F and E by the nut H, the latter being fixed to the centre piece. The lifting and lowering is done by the hand wheel J, which by worm and worm wheel turns shaft K and chain wheel L, and the sliding arm E is raised or depressed by the chain which is fastened to each end and tightly wound round wheel L. The third operation, the rotation of the arm F, is effected by turning the shaft N from the handle M, and, the motion passing through the change wheels is transferred to the worm wheel O, which is fixed rigidly on the central portion of the



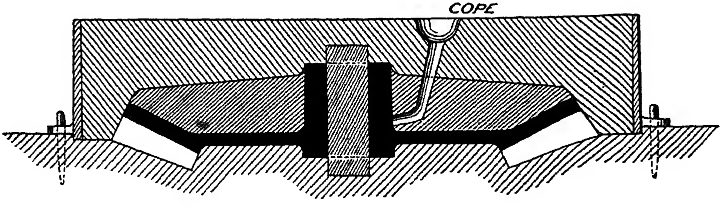


Fig. 48.
Bevel Wheel.

machine. Varying change wheels can be inserted in much the same way as in a screw-cutting lathe, and the revolutions or fractional parts made by the handle can be seen by means of the graduated disc P, so that any part of a circumference, such as the pitch of the teeth, can be accurately traversed by the mechanism last explained. The teeth are then formed in the sand after the

proper radius of arm is fixed, by lowering the tooth pattern, filling up with sand, raising, rotating the amount of the pitch, lowering again, and so on until the whole wheel is formed. The machine is now taken away by attaching the crane chain to the eye-bolt at the top of central spindle (*see also bottom of p.* 58).

Core boxes are needed for the wheel boss and segments between the arms, and for central hole. Wood strips are used as gauges to fix the cores in position and to preserve the proper thickness of the rim and arms; and a cope being placed over all, gates are formed and the wheel cast.

Bevel wheels are moulded in a similar manner to the above, the principal alterations being the strickle boards and tooth patterns. Fig. 48 will make this clear. A board A strikes out the back of the wheel in green sand, and parting sand is applied; an impression of the wheel back is then taken in the top box, and the latter removed to finish; board B next forms bottom face, cores and boss pattern completing the remainder. The core-boxes for the wheel arms are shewn in 72*b*, p. 61, and similar boxes will be found in Fig. 42.*

√ **Chilled Castings.**—Where a very hard and durable surface is required to a casting, "chilling" is effected by making that part of the mould, where the said face occurs, of iron. When the molten metal meets the surface of cold iron, it cools rapidly and forms crystals of white cast iron, hard yet brittle, where it meets the iron mould, and for a depth of an inch or more within the casting, according to the mixture used, or the weight of the chill mould; the rest of the casting is still grey and soft.

It would seem that the graphite crystals do not, under such circumstances, have time to form, and so the carbon becomes combined with the iron. Suitable objects for chill casting are:—rolls for plate mills used in forging plates (*see Fig.* 275, *p.* 280), and which require a great depth of chilling, as they have to be trued up again on being worn down; tram-car wheels, the rim only being chilled, this class of traffic not being capable of supporting the expense of steel tyres; points of plough-shares, which wear away at a very rapid rate in the earth; bushes for ordinary cart axles; railway chairs; pro-

* For other moulding machines see Appendices II., p. 785, and V., p. 969.

jectiles for large guns; and, in fact, all classes of work required to stand wear and tear, and not especially needing machining.

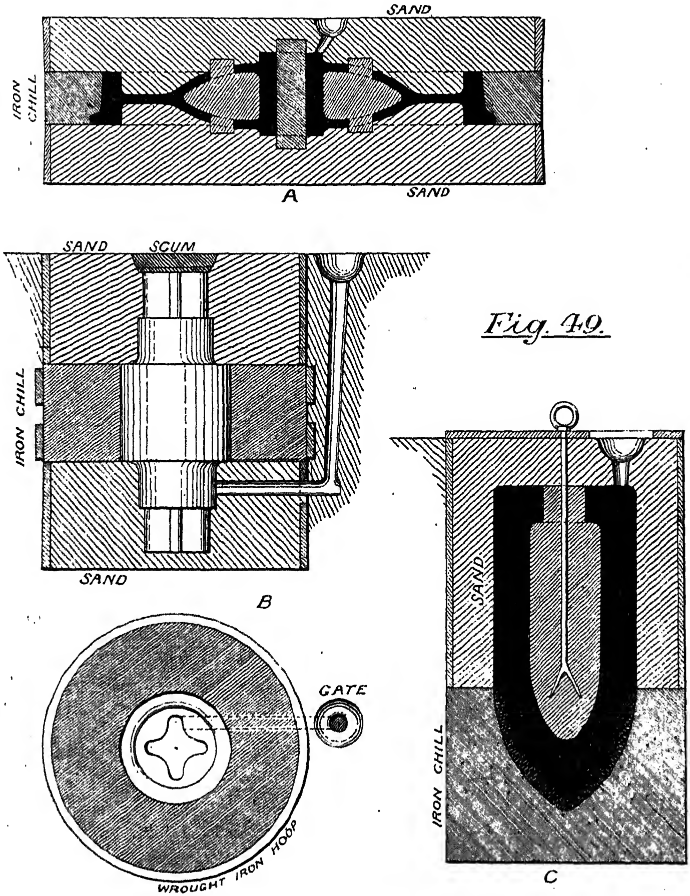


Fig. 49.

Chill Moulding.

Fig. 49 will explain the methods of chilling:—A, a car wheel rim; B, a plate roll; and C, a shell. Rolls and bushes are moulded upright, and share points made in an entire mould

of cast iron. The iron mould must be painted with a thin coating of very fine blackwash before casting, and some care must be taken in the forming of the gates, as, if there should not be sufficient pressure from the 'head' of metal used, the iron will recoil on meeting the cold mould, and form a rough casting. Care must also be used in the case of bushes, to remove the core chill before the casting cools down firmly upon it. The chills (the name given to the iron moulds) are usually made of good cast iron, though, in some rare instances, wrought iron has been used. Some of the details of chill moulding vary according to different authorities. Some founders purposely rust their chills on being first made, to assist the blackening in resisting the action of the metal, it being generally believed that the latter tends to fuse and injure the chill. Other founders, notably in the case of projectiles, neither rust them nor use blackening. The chills should be warmed before casting, in order to expel moisture, and should have a weight about six times the casting they are to chill, or the chilling will be too slow.

✓ **Malleable Castings** are obtained by taking the article, after being cast and cleaned up (this last is very important), and putting it, along with others, in an annealing furnace, in company with some substance that will absorb the carbon from the cast iron. Such substances are, oxide of iron in the form of scales from the rolling-mill, or some other of the metallic oxides, placed in the furnace in a state of powder. The intensity of the heat, and the time the casting should remain in the furnace, both depend on the size of the casting and the amount of malleability required, the usual rule being to keep it at a white heat for about a week, adding to this the time required to raise the temperature and to cool down.

Fig. 49*a* shows an annealing furnace with cast iron boxes A A holding the castings, which are covered with a layer of sand. Of course, it must be understood that it is only to a short depth below the surface that the casting becomes converted into wrought iron, though right through for small articles.

✓ **Softening.**—If a casting is so hard that it cannot be machined, it may be softened by heating and cooling out in common sand, or any other bad conductor of heat.

Brass Founding.—The moulds used in the casting of brass require no new description, the only difference in this class of work being the manner in which the metal is melted.

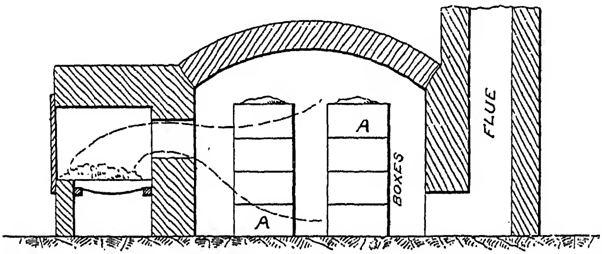


Fig. 49a.

Furnace for annealing Malleable castings.

As the castings required are much smaller than those we have recently been describing, the brass is made directly in crucibles of some impermeable material, black-lead being the best. The melting furnace is shewn in Fig. 50, and usually there are several of these side by side and separately connected with the chimney. The top of the furnace is only a little above the floor level, and in brass foundries it is customary to have the part of the shop near the furnace entirely reserved for casting purposes; the casting and moulding shops being entirely divided by a wall in the best establishments. The principal difficulty in the making of brass is that of the different fusing points of the two metals used—**Copper** and **Zinc**. Thus, copper melts at 1996° F., while the melting point of zinc is as low as 773° F.

The copper is first melted, and the zinc is only introduced a short time before casting, by means of tongs, pushing it down in small pieces under the melted copper. It should flare up on doing this, which is a sign that the heat is quick enough. If it is left in too long, much of the zinc will be lost by evaporation.*

* See also Appendix I., p. 747.

In bringing this chapter on casting and moulding to a close, a few practical points will be mentioned, although it should be clearly understood that perfect practice can only be obtained by actual work on such articles as have been mentioned in the text.

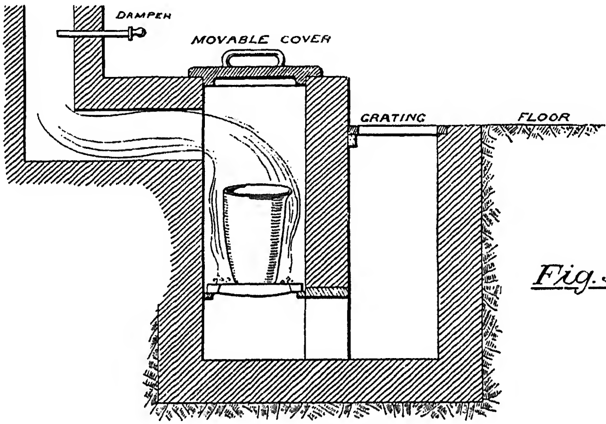


Fig. 50.

Crucible Furnace for Brass.

The Size of Gates, and the number of them, can only be determined by constantly watching the results obtained from previous work.

Flat gates ▬ should be avoided as much as possible, as they tend to clog, though sometimes they are beneficial when they are required to break off easily without damaging the casting. Fig. 51 is intended to represent diagrammatically the different gates and channels used to supply a mould; the pouring gate; the skimming gate, for the purpose of retaining the scum (and here some ingenuity is required to keep the latter in the skimming gate by centrifugal action, the whirling being produced by admitting the metal at a tangent); the sprues or connexions from skimming gate to mould (they may be of any number considered necessary); the feeding gate or gates, the use of which is to fill up any part of the casting which is likely to shrink; and the risers, which are to allow the whole of the air in the mould to pass out and so prevent blow-holes, the

soundness of the casting being also in the hands of the pourer, as he may keep the riser covered for a shorter or longer time.

The size of gates is determined by the fact that the metal must neither flow too slowly so as to choke, nor too quickly so as to break the mould. All the while the pouring is going on, the moulder agitates the metal in the gates by means of an iron rod, which he moves up and down until the metal has cooled so far as to prevent him doing so any longer; in this manner homogeneous casting is more likely to result.

A great deal of art lies in the ramming of the mould, but as a rule, the deeper portions of it should be rammed most, as there will be a greater head of metal on them. The floor of the shop should be well vented by a bed of coke and the insertion of pipes, to take away the gases from all work done on the floor, and coke dust should be put in the joints of loam building; some cores too should have a large amount of coke in their centre.

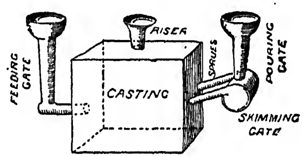


Fig. 51.
Diagram of Gates.

The venting of the mould is also a matter which requires a great deal of practical experience to enable it to be done with success. Large cores, enclosed on three sides by metal ('pockets,' moulders call them), should be particularly well pierced, and green sand moulds should be much better vented than loam or dry sand work, on account of the steam rising from the damp sand and the compactness of the latter.

Cores require good support by means of plates or wires, especially such as those in Figs. 25, 34, and 43, and all cores that are not held down by the shape of the mould, should be so fastened, for they are so much lighter than the molten metal, that they would float out of position if left to themselves.

Cores should be dried in the stove for about twelve hours, and should only be placed in the mould a short time before pouring, to prevent the absorption of moisture by them from the mould.

Patterns are lifted out of the sand by screwing rods or handles into them (*see Fig.* 73*d*, *p.* 67), and raising slowly, at the

same time rapping the pattern carefully to prevent the sand adhering, and a few points should here be noticed as regards the finishing of the mould. The moulder uses trowels, and 'sleekers,' (which last are only trowels of special shapes) to smooth away any broken portion, but, if the mould is made too smooth, there is great danger of blistering or scabbing, from the fact that the mould, having lost to some extent its porosity, refuses to allow the escape of gas, and it is generally understood by moulders that the hand makes the best trowel, though certainly it is always better to let a mould remain, if possible, just as the pattern left it.

In Fig. 52, a few moulders' tools are sketched.

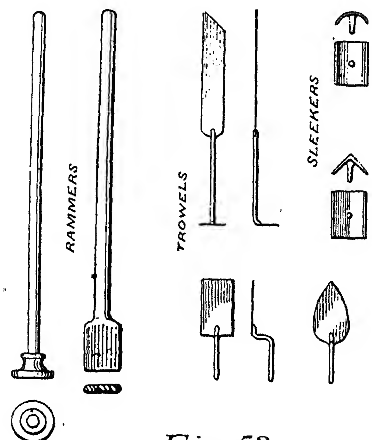


Fig. 52.
Moulders' Tools.

The upper box is usually termed the '*cope*,' which also applies to the outer mould, and the lower box is sometimes called the '*drag*.' The cope should be well weighted to ensure sound castings. When two boxes are used, they fit together as shewn in Fig. 2, and can be easily replaced, but if the floor serves as bottom box, exact correspondence is obtained by driving stakes into the ground through the lugs of the top box.

Chaplets have been before mentioned, and are required to stay cores that cannot be otherwise supported; their use, however, is not advisable, for they tend to produce weakness in the casting in which they remain. They should be tinned, or at least freed from rust, to ensure uniting with the casting.

We have already described the charging of the cupola; it remains for us to explain the method of tapping it. The man at the cupola is provided with two iron rods; one he uses to pierce a hole in the clay stopping, which he does as soon as the moulder

has brought his ladle under the mouth of the cupola; the other rod has a flat end, with a lump of clay adhering. As soon as the ladle is full, he applies the clay to the mouth in order to stop the flow. To tap with safety, he should stand on one side of the trough and use his tapping rod obliquely, while on stopping again, he should cut off the stream from the top side.

Mixtures of Iron.—This is another art which nothing but observation and practical experience can reduce to a nicety, different mixtures being used for the same purpose by different founders; indeed, the success of certain firms depends in a great measure on the mixtures used. Still, a good idea can be given of what is required for each purpose.

The varieties of pig iron having been already stated, we will now consider each separately.

No. 1 is the weakest but most fluid of all the pigs, and may be used by itself for ornamental castings on account of the ease with which it fills the corners of the mould, but it is usual to mix it with '*scrap*' to increase its hardness, ultimate strength, and closeness of grain.

No. 2 is finer in grain and stronger than No. 1, and is used wherever some strength is required with great fluidity.

No. 3 combines the greatest degree of strength consistent with fluidity, and is therefore most extensively used, and in great favour with founders.

No. 4 is the strongest pig for foundry use (the remaining numbers, 5 and upward, being only required for conversion into wrought iron), and is therefore used for heavy castings requiring strength, such as girders, columns, bed plates, &c.

For strong castings two-thirds of No. 4 may also be used with one-third of No. 1.

Scrap is the name given to the broken up parts of old castings, which of course may be divided into good and bad scrap. Some founders place great reliance on it, using nothing but scrap with an admixture of No. 1, say two-thirds of scrap to one-third of No. 1, while others prefer using an iron like No. 3, mixing with it only a little scrap to strengthen it, and so produce a harder, close-grained casting. It is also a good plan to mix iron from different blasts.

While speaking of scrap, we cannot do better than endeavour to understand the advantage or otherwise of remelting. We have before said that remelting is a disadvantage. It is true that the iron becomes purer as regards the elimination of graphite, acquiring a *whiter* appearance, with, at the same time, increased strength and closeness of grain; but, on account of other impurities, it is no longer as tough as before, and its ultimate extension is therefore decreased. Unwin says: 'Remelting improves the strength, but if repeated too long the tensile and transverse strengths suffer, though the crushing strength and hardness increase.' (*See Appendix II., p.* 779.)

For chilled castings, a strong iron, as Nos. 3 and 4, is needed, because the chilling weakens the metal.

Malleable castings require a pure mottled iron, or at least one having very little grey mixed with it; for if the particles of graphite present in coarse grey iron are taken away in the furnace, honeycombing will result.

Girders and columns must have a strong and elastic mixture; cylinders should be treated for hardness as well as strength, and therefore require as much white iron as convenient; pulleys need a soft mixture, such as a large proportion of No. 1 with a little of No. 3 for strength.

Steel Casting requires no explanation. Its conversion from iron will be treated in a subsequent chapter. The only difference in the foundry is, that in order to prevent 'honeycombing,' which has been a great trouble ever since steel castings were first used, great care has to be exercised, and even then many castings are wasted, while brittleness is only prevented by slow annealing for over a week or a fortnight of time. (*See Appendix I., p.* 747.)

Sir Joseph Whitworth introduced a method of compressing the steel while in the molten ingot by powerful hydraulic pressure, in order to prevent this troublesome honeycombing, the only objection to his process being considerably increased cost. (*See also bottom of p.* 82, *and Appendix III., p.* 790.)

CHAPTER II.

PATTERN MAKING AND CASTING DESIGN.

The arts of moulding and pattern making are so inevitably interwoven, the pattern maker especially requiring to know at least the *principles* of moulding, that it would be quite impossible to make these two first chapters independent of each other, and as the subject of pattern-making has been so much entered on in our last there is little left to say as regards the *forms* which patterns should take; but whatever has been omitted we will endeavour now to supply.

Wood.—Pine and mahogany are the two woods most extensively used, and are kept in large stores until they are perfectly dry. White and yellow deal are used for the larger and less accurate patterns, but are not hard enough for the better ones, nor so good to resist warping as other woods.

Mahogany warps and shrinks very little in drying, and is consequently in great favour. On account of that fact, and the ease with which it is worked, it is considered the best wood for pattern making; expense, however, precludes its use for large patterns.

Cherry wood ranks next to mahogany; it warps a little more, and is rather more troublesome to work.

Sycamore, lime tree, and American walnut are other woods used.

Iron patterns are employed for lighter objects, or those requiring very great accuracy.

It would be well before proceeding further to consider the way in which timber warps in drying, as this will show us some of the reasons for building up a pattern in a particular way.

The shrinkage of wood in length is so inconsiderable that it may be disregarded; and, in fact, some imagine there is none.

The greatest contraction is across the grain. To prevent the splitting of the tree, as at Fig. 53, it is sawn up into planks whilst green, and these in drying take the form sketched in Fig. 54. The outside layers contract the most, these being the youngest

wood and, as a consequence, the centre plank is narrower at the edges while the side planks become drawn in, in the manner shewn at A. If the tree be cut into quarters, each quarter will contract, according to the above law, in the manner sketched at Fig. 55, the sector piece at B becoming narrower, the square at C becoming rhomboidal, and the circle at D elliptical. Even after

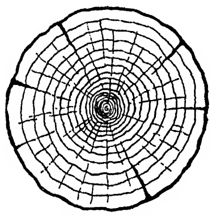


Fig. 53.

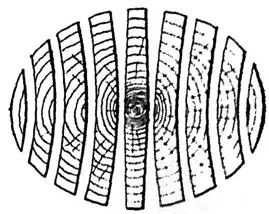


Fig. 54.

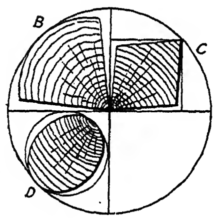


Fig. 55.

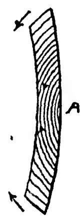


Fig. 56.

Shrinkage of Wood.

timber has been thoroughly dried, it will always warp if a good shaving be taken off it, as may be seen at Fig. 56, for a moist surface is exposed, which, drying, must contract the side that has been planed. It is, therefore, suggested by some authorities that, in first beginning a pattern, the wood should be marked out and cut to the size required, so as to have some little time to set before being used. (*See Appendix II., p.* 787.)

Tools and appliances.—It would be unnecessary in a work like the present to occupy the student with elaborate descriptions of the hand tools used by the pattern maker. The following illustrated list will serve to explain them :

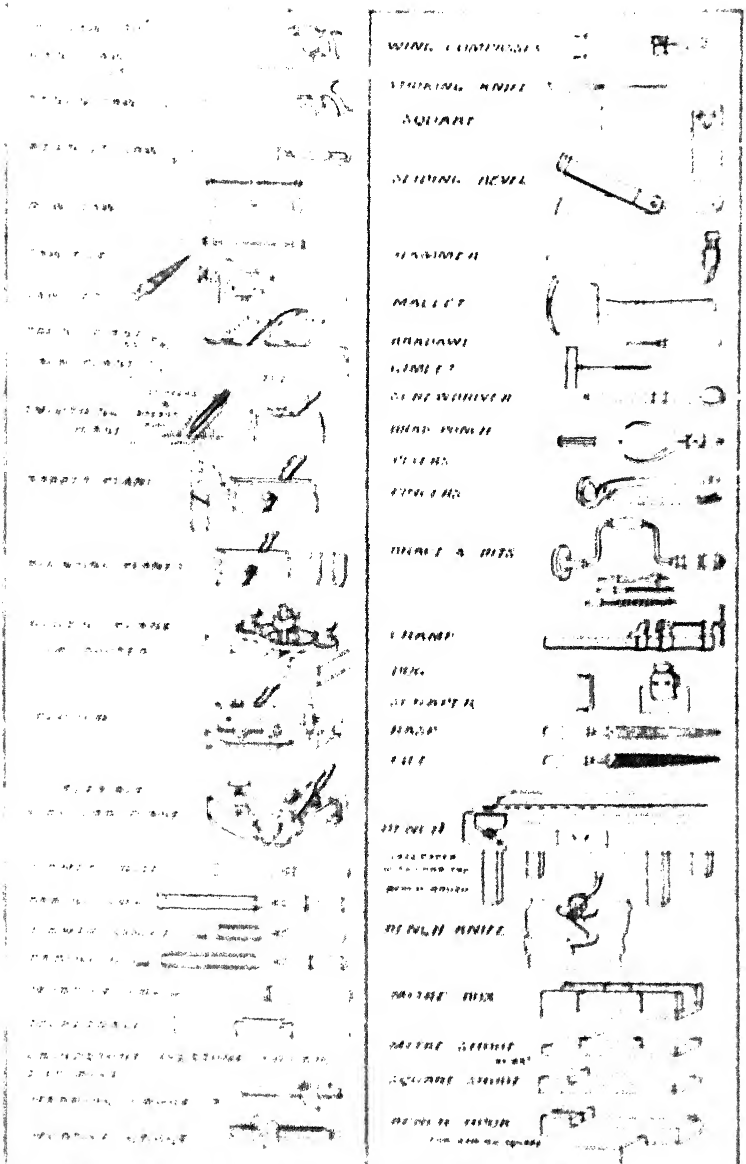


Fig 32 Tools and Appliances

The flexible plane is very useful for truing large concave or convex surfaces, a symmetrical curvature being given to the sole by simultaneous adjustment at front and back.

Of machine tools, two or three lathes are required; the first, with a long bed and two headstocks, Fig. 59; the second, a large face lathe for turning wheel rims, Fig. 60; and the third, a light face lathe for small articles, Fig. 61. Their speeds must be considerably above those used for iron turning. A tool rest is used to them all, and the work is done entirely by hand, the tools necessary having such edges as are sketched at Fig. 62.

The mandrel of the lathe is provided with a chuck, which has a different form for each lathe; thus, the face lathe has a screw on the mandrel to receive the flange chuck (*see* A, Fig. 60), and the face plate on which the pattern is turned is supplied by a disc of wood bolted to the chuck. The lathe with the two headstocks is provided with a chuck of the form shewn at B, Fig. 59, which is well pressed into the end of the pattern, and so compels the latter to turn with the mandrel; and in the small lathe, the pattern is screwed on the mandrel at C.

Other machine tools required are:—a band saw, circular saw, and, if possible, a wood-planing machine.

Arrangement of Mould.—It is the duty of the pattern maker to so arrange the moulding of his pattern as to cheapen it (the moulding) to the utmost, and give the least trouble in the foundry. Thus, if the mould is to be in green sand, as little dried core work is to be used as possible, and very often a great deal may be done by the introduction of *loose pieces*, which are left in the sand after the body of the pattern has been withdrawn, and are then removed in another direction (*see* A, Fig. 30). He should put as little of the pattern in the top box as is practicable, for it is evident that this part of the mould must receive the worst treatment, being lifted off and turned over, perhaps more than once. Added to this, the fact that the cope has to be taken away so as to leave the pattern behind, means the using of a good deal of care, despite which much broken sand may still appear; while the half pattern in the bottom box may be lifted in full daylight, and no accident need happen to it. We may here mention generally the method of withdrawing trouble-

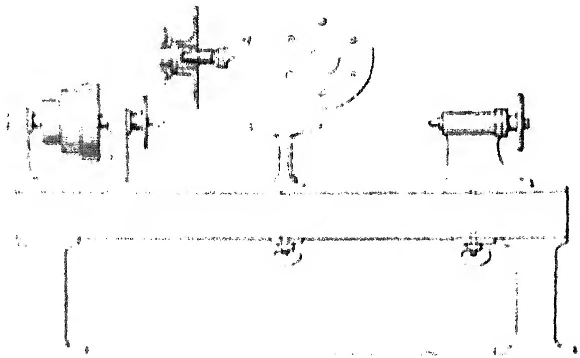


Fig. 59.

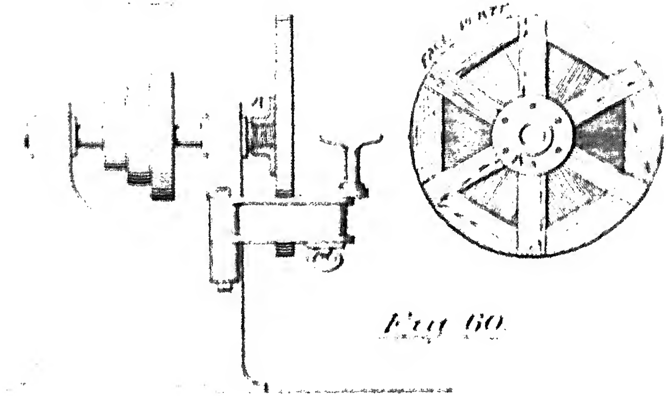


Fig. 60.

Fig. 61.

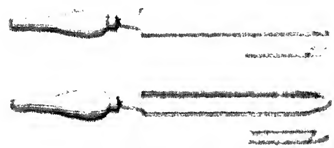


Fig. 62.

Wood-turning Lathes.

some parts of a pattern by lifting portions of sand on
'*drawbacks.*' An example often quoted, and a very goo
purposes of explanation, is that of a lathe bed.

Fig. 62*a* shews a section of the pattern lying in th
The bed is reversed, so that the planed surfaces shall be

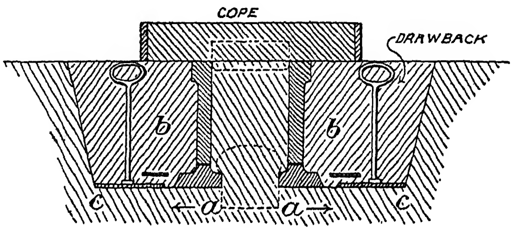


Fig. 62*a*.
Lathe Bed.

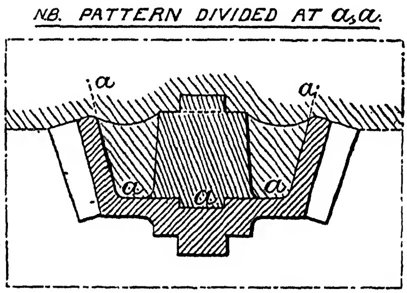


Fig. 62 *b*.
Bevel Pinion.

scum; and these surfaces are in the pattern made loose pie
The upper portion of the pattern can be easily removed, b
problem is to withdraw the pieces *a a*. This can only be ac
by building the sand at *b b* on plates *c c*, provided with hand
which they and the sand upon them may be removed, aft

upper portion of the pattern has been taken away. The pieces *a a* may now be withdrawn in the direction of the arrows.

Of course the middle space in the pattern might have been taken out by means of cores, as suggested by dotted lines, and this example has been introduced to shew how necessary it is for the pattern maker to carefully consider the best way of moulding the object in hand before commencing to make his pattern.

A difficult case may always be overcome by the use of either loose pieces or drawbacks, or by coreing, but if there should be too much of the former to do, the latter had better be resorted to. In the case of the pulley in Fig. [illegible] we have [illegible] but difficult moulding, and many would prefer to have a print in the pattern, and a core box from which to mould the hollow.

While speaking of loose pieces it may be mentioned that for safety of withdrawal some portion of the pattern is often left loosely attached, although not apparently necessary. In Fig. [illegible] the boss and arms of the bevel pinion are loose, and are taken away with the top box. The two portions of the pattern may then be withdrawn without risk of breaking away the narrow portions of sand *a a*.

Pattern Building. The pattern is first pencilled out, full size, on chalked boards, and with sufficient 'views' to make the subject clear. Core prints, core boxes, &c., are also supplied in the drawing, and taper given to all faces parallel to direction of withdrawal.

All surfaces to be machined should have an allowance of one eighth of an inch for iron or steel, and for brass one sixteenth of an inch. These amounts vary somewhat in special cases, as for example in a long bed plate, where three-eighths of an inch or more may be required on account of the probable warping of the casting. An upward camber of one eighth of an inch per six feet is given in patterns for lathe beds.

In small cylinders one-eighth of an inch all round the bore would be sufficient, but large cylinders would need a quarter of an inch.

Some ingenuity has to be used in the building of patterns. If they are carved out of the solid they have more chance of warping, for

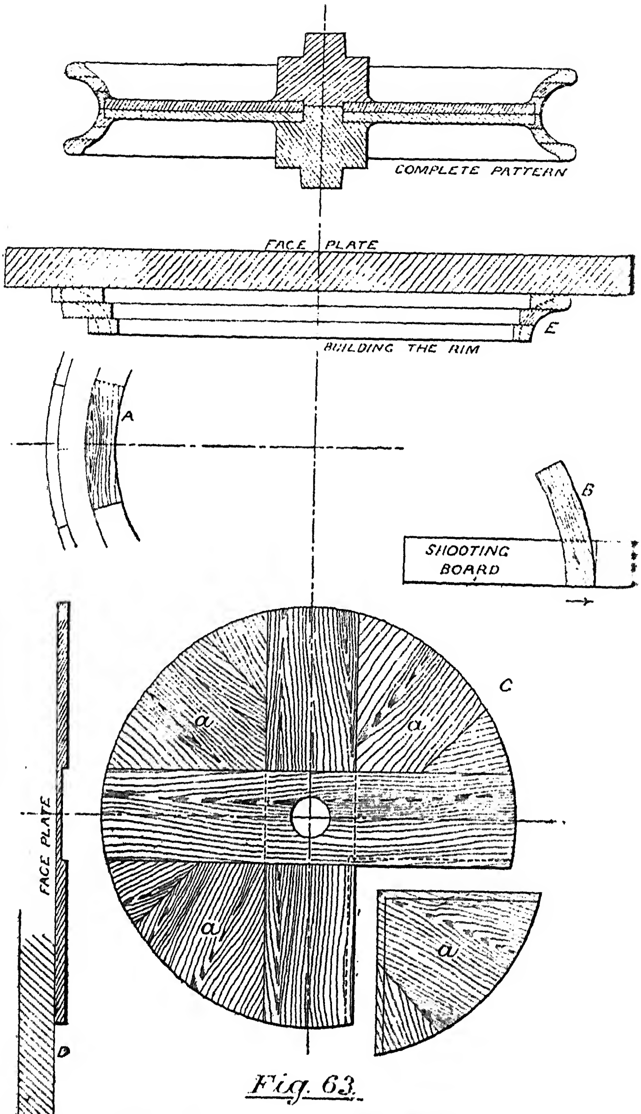


Fig. 63.

Making a Pulley Pattern.

reasons which have been previously mentioned, and so, patterns of the larger kind at least, are made of layers well glued together. Fig. 63 represents the making of the pattern for the pulley shewn in Fig. 10. The rim is first built on the face plate of the lathe in the following manner:—

Pieces of wood of the form shewn at A being sawn to shape, are truly planed on one side at least, and also at the ends, by the help of a shooting board B, which is used as a guide; they are then fitted together to form the first layers of the rim, by glueing to the face plate, taking care to put a *strip of paper between*, which is always done when work is to be afterwards removed.

When dry, this layer is turned to a true plane, and another superposed in a similar manner, but so as to 'break joint,' and so on; the whole is lastly turned on the face E, and, being carefully removed, is reversed, and again turned on the back side. So much for the rim. The plate of the wheel is next formed, as at C, by halving one plank over another at right angles, these being grooved to receive the filling quarters *a a;* the plate is next bored at the centre to receive the boss, and turned on the outside (*see* D). A rabbet having been formed in the rim to receive the plate, one half of the pattern is complete.

Fig. 64 shews the halving for a pulley of six arms, each batten being cut to a fraction of its depth indicated by the figures; and Fig. 65 shows the method used for five arms, the boss being required for fastening purposes. These are for the arms themselves.

The upright ribs, used to strengthen such arms (*see* Fig. 12), would be halved on their narrow edges, and the boss filled in by segments as at E, Fig. 65.

Pipes.—Patterns for small pipes are made out of the solid, but those of a large size are built up as in Fig. 65*a*. Polygonal half discs are placed at either end, and at intervals. Upon these are carefully fastened with glue (and screws if necessary) the pieces forming the rim. The two surfaces A and B being made true planes, the half pipes are glued together with paper between, and, having dogs driven in at the ends C, are centred in the lathe, Fig. 59. If flanges are needed the ends would be turned as in Fig. 66, and the flanges fitted in the manner shewn.

Bent pipes cannot be built in this way, on account of the

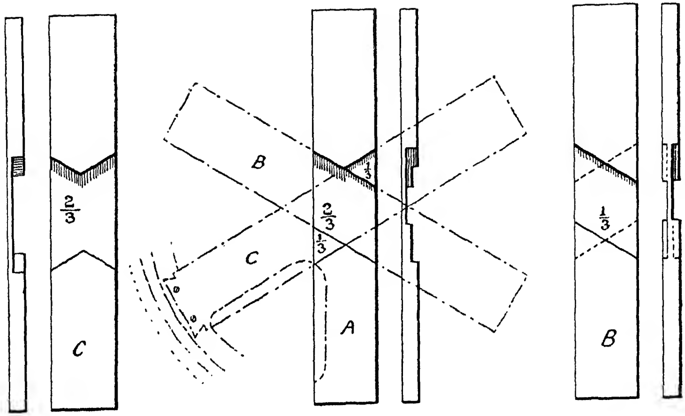


Fig. 64

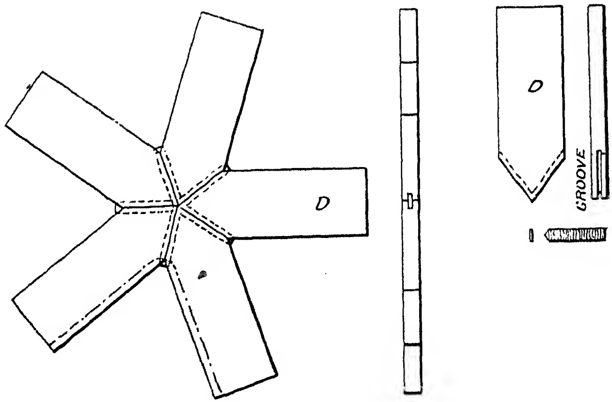
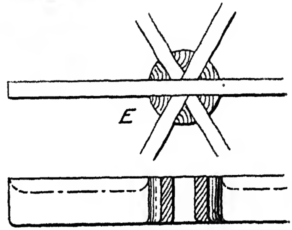


Fig. 65.



Methods of joining Wheel arms.

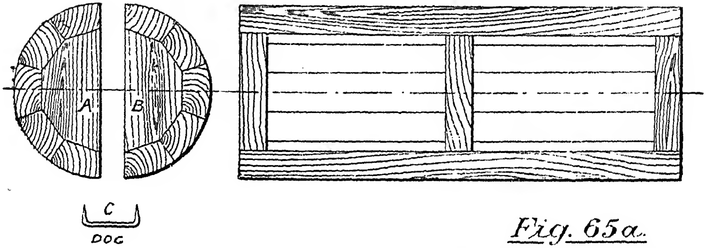


Fig. 65a.
Straight Pipe.

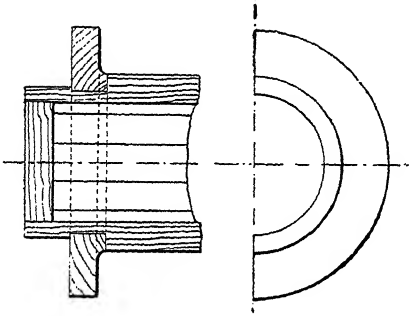


Fig. 66.

Fixing flanges.

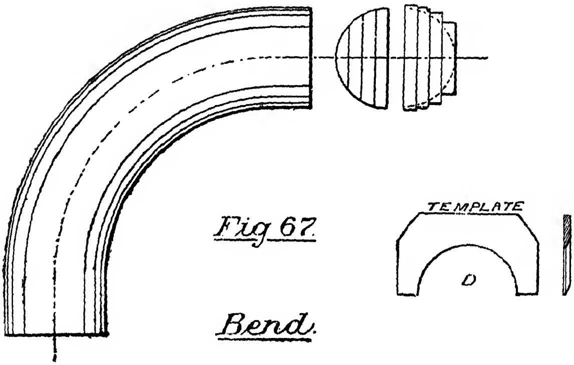


Fig 67

Bend.

impossibility of bending each lath of wood; they are therefore made as in Fig. 67, which shews a plan and end view. In the example shewn, the pipe must be worked out by gouge and spoke-

shave, and tried from time to time by applying the template D; in fact, working in wood much as the moulder works in loam at Fig. 25 in the last chapter.

A still more handy way of making quarter bends is to turn a built-up ring of semi-circular section on the face plate of the lathe, Fig. 60, and afterwards to cut into four equal pieces, as shewn in dotted lines, Fig. 67A. On removing and placing back to back,

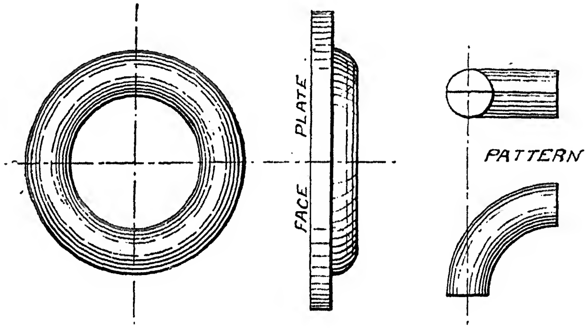


Fig. 67a.

Quarter Bends.

we have clearly two complete patterns, and flanges may be added as necessary.

If a single bent pipe—of whatever form of bend (so that it be in one plane)—is required, it should be moulded as at Figs. 25, 26, or 26*a*; the pattern maker will supply the necessary templates. Curiously shaped pipes, of varying bore, may have a core box only; the outer pattern being built by the moulder, who lays thickness pieces on the core, smoothing off with loam; after taking an impression, the thickness pieces are removed (*see* description to Figs. 25 and 26).

Joints.—Many other methods of jointing, besides halving and rabbeting, are, of course, used, such as dovetailing and tenoning, but we must content ourselves with a general notion of the making of a pattern.

Varnishing.—When finished and sand-papered, the pattern

is carefully varnished so as to preserve it from moisture, and present a smoother surface to the mould.

Core Boxes.—The simplest kinds are such as are shewn at Figs. 8, 10, and 14, where half the core lies in each box. Pegs to unite them are formed by knocking rough rods of wood through the steel plates, Fig. 68, and then driving these into holes in the core boxes (see *a*, Fig. 69). The pegs need not have more than a quarter or five-sixteenths of-an-inch projection, as, if they are longer, they may stick. The exact correspondence of peg and socket is found by pressing some little object, such as a pinhead, between the boxes, and using these marks as centres. Pegs are also required to unite the halves of patterns. Wooden pegs are now greatly superseded by brass dowels (Fig. 68*a*).

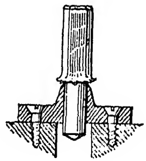


Fig. 68.
Making pegs.

The hollows of cylindrical core boxes are gauged by the use of a property of the semi-circle—viz., that the angle contained by it is always a right angle. So that the box may be gouged out as in Fig. 69, and tried from time to time with a set square.

More complicated core boxes have been already drawn at Figs. 30, 31, 33, and 41. The last one may be noticed as a case of a box that must be loose on every side in order to effect the safe removal of the core.

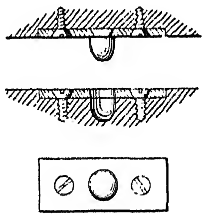


Fig. 68a.
Brass dowels.

Core Prints.—At this point we may consider shortly the different kinds of core prints.

Simple cylindrical core prints are shewn at Figs. 10, 15, 18, 37 : they require a slight taper in direction of withdrawal. Sometimes it is necessary for economy to core the bolt holes in the flange of a steam pipe, especially if the holes are to be square A little consideration here of direction of withdrawal will show that, if we used prints, they would need to be of a very special kind, so they are usually dispensed with altogether,

and a template given to the moulder of the shape shewn at Fig. 69*a*.

The cores, of a length equal to the thickness (full) of the flange, are pushed down to their place, and held there by friction.

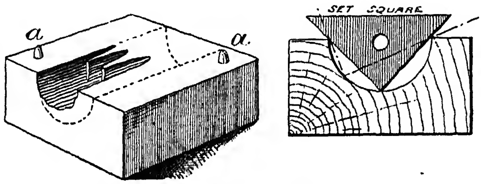


Fig. 69.
Gouging Core-box.

But a case similar to the above might occur, when, on account of the weight of the core or the accuracy required, it might be advisable to have prints, and as plain cylindrical ones would not

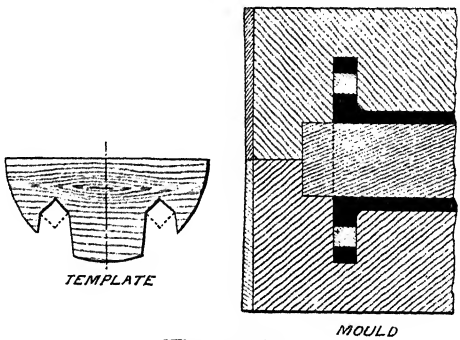


Fig. 69a.
Bolt-hole cores.

draw, as previously stated, we are obliged to have recourse to the 'tail' prints in Fig. 69*b*. Here A represents the pattern with its prints, B is the core box for the hole, and C represents the finished mould. The portions *b b* are to be filled in by the moulder either by hand, or in the case of a difficult shape, by cores made from boxes.

Yet another form of print is required, where the core can be supported at one end only. That part of the core lying in the print

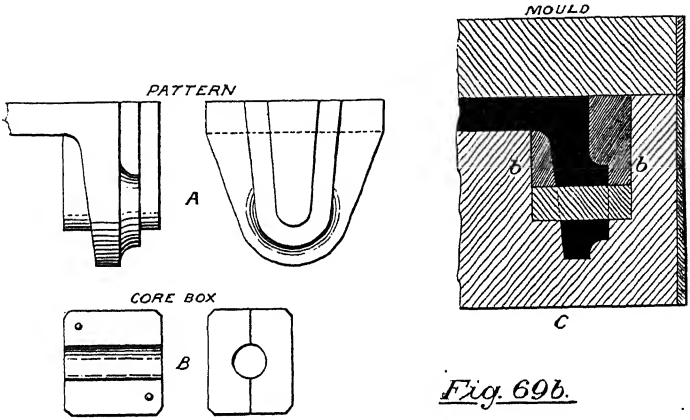


Fig. 69b.

Tail Prints.

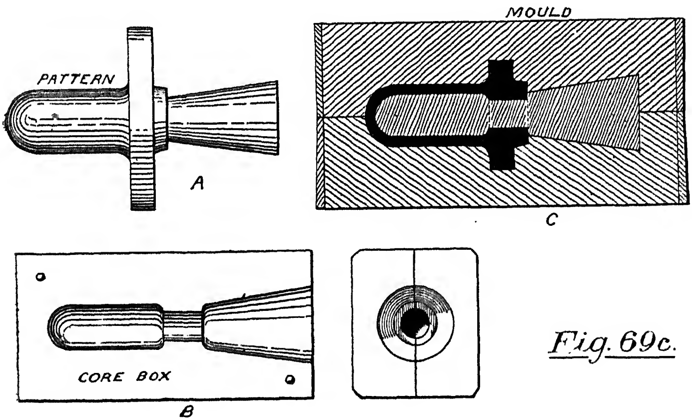


Fig. 69c.

Dummy Gland with Balance Print.

matrix has then the whole weight of the core to support, and must in consequence be large enough for balance. Fig. 69*c* will explain

what is meant: where A is the pattern, B the core box, and C the mould, the object being a 'dummy' gland for a steam cylinder.

Referring again to the **Worm Wheel** in Fig. 12, the method of making the pattern will be understood by the help of Figs. 70, 71, and 72. It may be built in the way shewn for the pulley in Fig. 63, and, after being turned on the rim, blocks of hard wood are fitted on each half of the pattern, and glued in the manner suggested at D (Fig. 72).

The outside surface of these blocks is now turned so as to give a solid rim of wood, from which the teeth are to be cut. To do this a stud A, Fig. 71 (on the table of a wood-working machine) is fixed at the angle of the worm thread, and the wheel pattern set upon it, so that it can be rotated carefully the amount of the pitch, by gearing, much on the same principle as in a moulding machine. A revolving cutter B, driven at from 2,000 to 3,000 revolutions per minute, is advanced to the pattern, and cuts out the space between the teeth; the diameter *d* of this cutter must be the same as that of the worm. When this operation is completed, the wheel is removed and placed on stud C, Fig. 72. The wrought-iron worm intended to work with the casting, being marked with red ochre, is now advanced, together with its wood bearings, to gear with the pattern, and the worm is rotated; then, wherever a little mark is left by contact of the worm, the wood is gouged away until a perfectly correct fit is obtained.

Spur Wheels too small to make by machine moulding may have their teeth formed by the revolving cutter shewn at B, but of course, in that case, the axes of wheel and cutter are at right angles.

For machine-moulded wheels, either spur or bevel, the moulder is to be supplied with a block of pine with two teeth dovetailed in harder wood, as in Fig. 72*a* (the machine is shewn at Fig. 46).

In both sketches the direction of withdrawal is shewn by the arrows, and it will be seen that, although the bevel teeth withdraw without difficulty, there would be some risk of the sand sticking to the pattern in the case of the spur teeth, which are made truly perpendicular and without taper. To avoid such an accident a finger bit A is provided, which, fitting in the hollow between the two teeth, presses down the sand as the block is lifted.

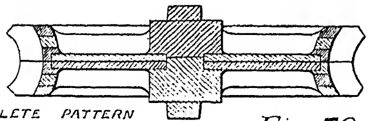


Fig. 70.

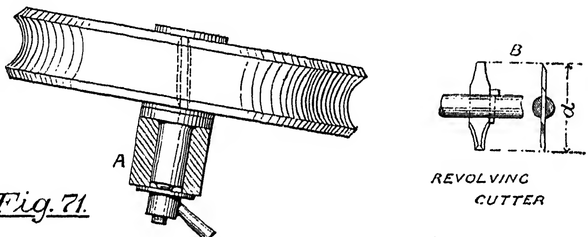


Fig. 71.

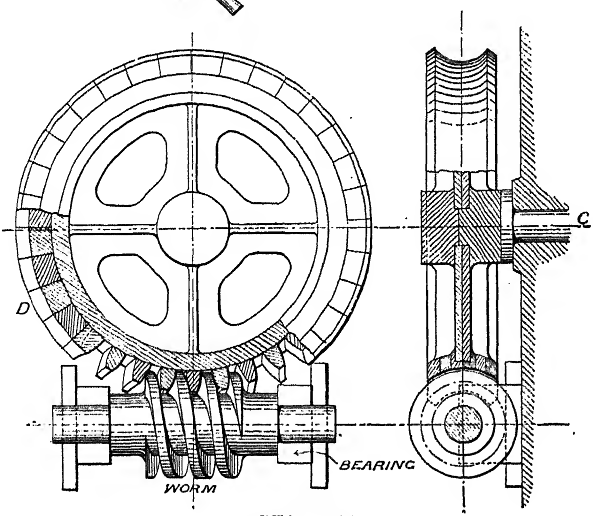


Fig. 72.

Worm-Wheel Pattern.

In Fig. 72*b* we have the core box required for the arms of a machine-moulded spur wheel; its description will serve also for bevel wheels. A represents the casting to be obtained, having six arms, and the box at B is so designed as to core out a space of one-sixth of that within the wheel rim. The box being in the

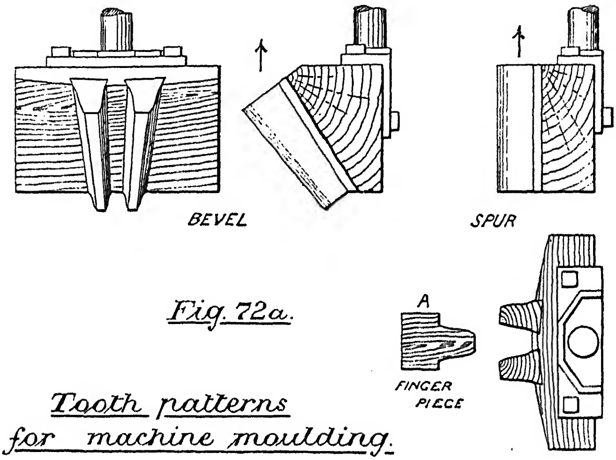


Fig. 72a.

Tooth patterns for machine moulding.

foundry is placed on a true table *a*, and after filling with loam, is smoothed off with straight-edged batten at *b*. Six of these cores being dried and blackwashed, a pattern for the boss of the wheel is now necessary to complete the mould.

Small **Bevel Pinions** require the patience of the pattern maker. Referring to Fig. 73, which is the section of a bevel pinion, it will be seen that the teeth vary in size from A to B, and must, therefore, be entirely gouged out by hand. The body of the pattern is carefully turned as at C C, while blocks D, for the teeth, are planted on in hard wood and again turned, as in the last example. The section of the tooth now being set out by compass or template at A and B, the teeth have to be cut out and finished by hand. The teeth at B are made correctly lineable with those at A by means of the wooden spindle E, carrying a straight edge F so cut as to be always truly radial when moved round the surface A B.

[illegible] form, such as worm wheels and helical wheels, [illegible] really be formed by templates at different planes [illegible] at top, bottom, and centre of tooth, but more will [illegible] of wheel teeth in Part II.

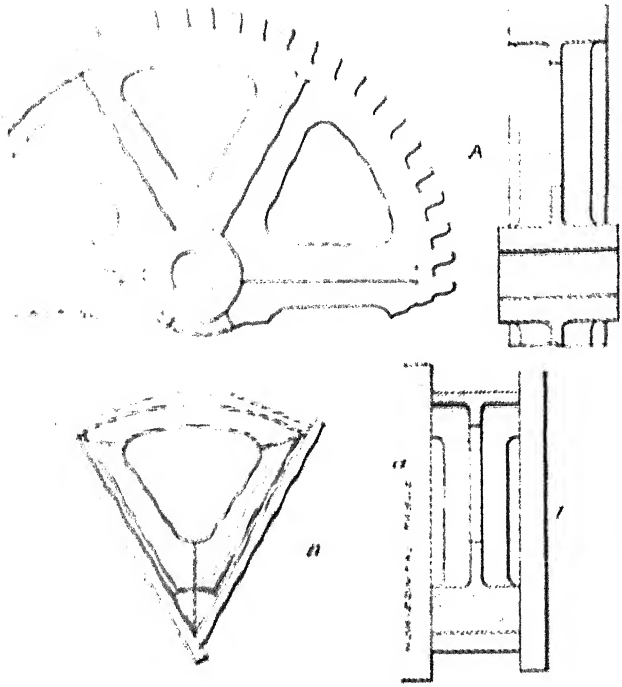


Fig. 72b.

Core boxes for wheel arms

Striking, stricking, sweeping, or **Loam Boards** — various names for the same object — are the only remaining patterns that need mention. They should be bevelled off at the striking edge, and their various forms can be readily grasped by reference to Chap. I.

Contraction of Castings.—This is a subject involving both thought and practice, and although a few general rules can be given, success depends on very many points. It has been previously mentioned that the moulder raps the rod that draws the pattern from the sand. This rapping taking place in a

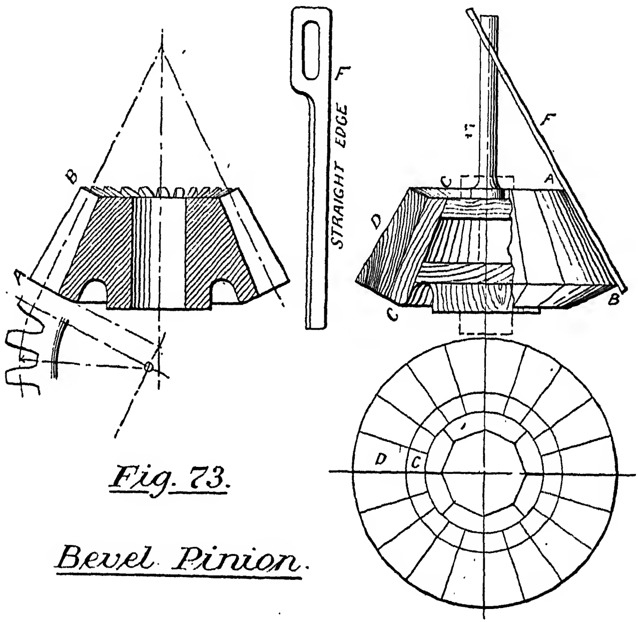


Fig. 73.

Bevel Pinion.

horizontal direction, it is evident that the sides of the mould only are affected by it.

The pattern maker must not only take account of this, but also of the particular moulder he has to deal with, for some moulders lift a pattern with less rapping than others. In small castings, up to about six inches across, the enlargement of the mould by rapping will be about compensated by the shrinkage of the casting; but in large moulds, the amount of shrinkage will be so much greater than the effect of rapping, that the latter may be entirely overlooked, account being taken only of the in-

crease in size of pattern necessary to compensate for contraction. Patterns two to three inches across, or less, should be made about $\frac{1}{32}''$ smaller to allow for rapping only, and as this does not take place in an upward or downward direction, there should always be full allowance for contraction at these places.

The greatest shrinkage due to cooling will usually occur where there is the greatest body of metal, and use must be made of this knowledge by the pattern maker.

The linear contraction for different metals is as follows:—

Cast Iron	$\frac{1}{8}''$	per foot	=	·125″
Brass	$\frac{3}{16}''$	,,	=	·187″
Gun Metal	$\frac{1}{6}''$	,,	=	·166″
Steel	$\frac{3}{16}''$	,,	=	·187″
Malleable Cast Iron	$\frac{3}{16}''$	,,	=	·187″
Aluminium	$\frac{17}{64}''$	,,	=	·265″

Spur wheels about 2 ft. 6 in. diameter, contract $\frac{1}{32}''$ per foot, and such wheels vary their contraction, increasing to $\frac{1}{10}''$ per foot for a wheel 10 ft. diameter (Box).

Three-foot rules, longer than the ordinary rules by the above fractions, are called 'Contraction Rules,' and are used by the pattern-makers, but with much care and judgment.

When wooden patterns are made, from which are to be moulded metal ones, a double contraction should be allowed on account of the two mouldings necessary to produce the required casting in the first case, and the consequent double shrinkage.

Metal Patterns are required for light work or when a great length of service is required. Such patterns are usually the same as the wooden ones from which they are made; but there are other examples of moulding with iron or brass patterns, as in **Plate Moulding.** This is handy for such small articles as occur in a brass foundry; Fig. 73*a* will shew the method. A wrought-iron plate *a* is provided with half patterns on either side, made in brass and carefully finished. Prints are run for connecting each pattern, so making channels for the flow of the metal. The plate also has corners *b b*, so that when put between the boxes *c c*, and rammed up with sand, exact correspondence of the boxes is obtained. Except for blackening and fixing of

pouring gate, &c., the mould is now complete, and will, no doubt, be admitted as economically made. Of course this method will serve only when a large number of castings are required of the same kind.

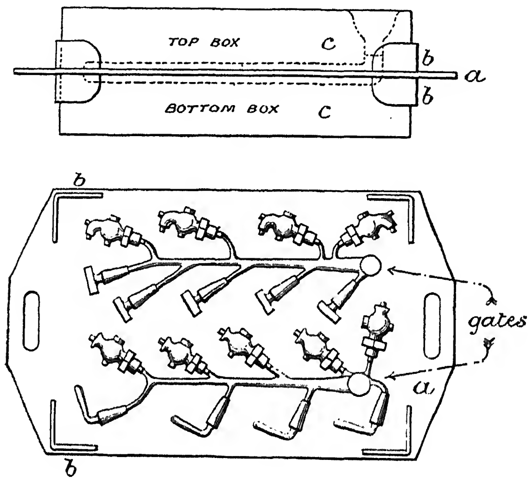


Fig. 73a.
Plate Moulding.

Stopping-off is a process which often serves to utilise a pattern temporarily for a slightly different purpose to that first intended, and so to effect economy. A simple example will suffice.

In Fig. 73*b*, a pipe pattern with flanges is shewn; we will suppose a shorter bend is required.

All that is necessary is to fix a flange on at A, and provide a stopping-off piece B of the same size as the flange, having a print attached for the core.

C represents a plan of the mould, with the stopping-off piece in position; the portion D being filled up by the moulder. The rest of the moulding will be easily understood.

In the propeller which we moulded in our last chapter (Fig. 35) a screw template was placed outside the mould. There are cases, however, when a screw is to be moulded in loam, but

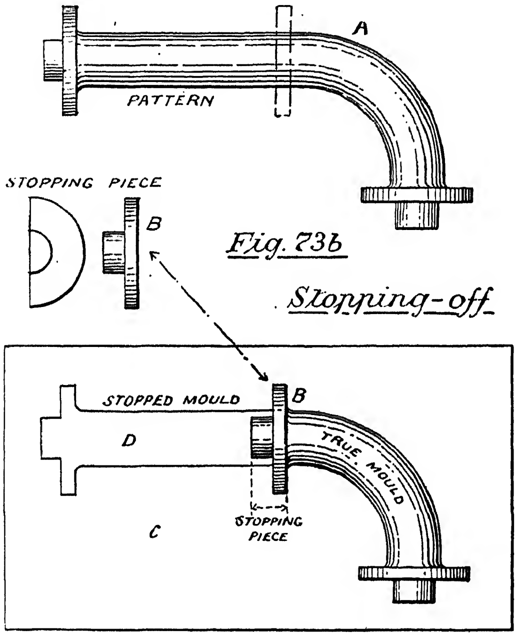


Fig. 73b
Stopping-off

where the course mentioned cannot, for certain reasons, be followed. Such is shewn at Fig. 73*c*.

A Chain Barrel for a crane is formed with a helical groove to receive the chain. A is the casting required; B shews the striking out of the loam, and C the finished mould. The only portion requiring explanation is the screw *d*, made of hardwood. It is fixed to the bottom plate, and has the same pitch as the chain groove, though, of course, a more abrupt rise (for this reason made as large as convenient). The striking board runs on the screw, being supported by a roller, and balanced by weight, as in the case of the propeller.

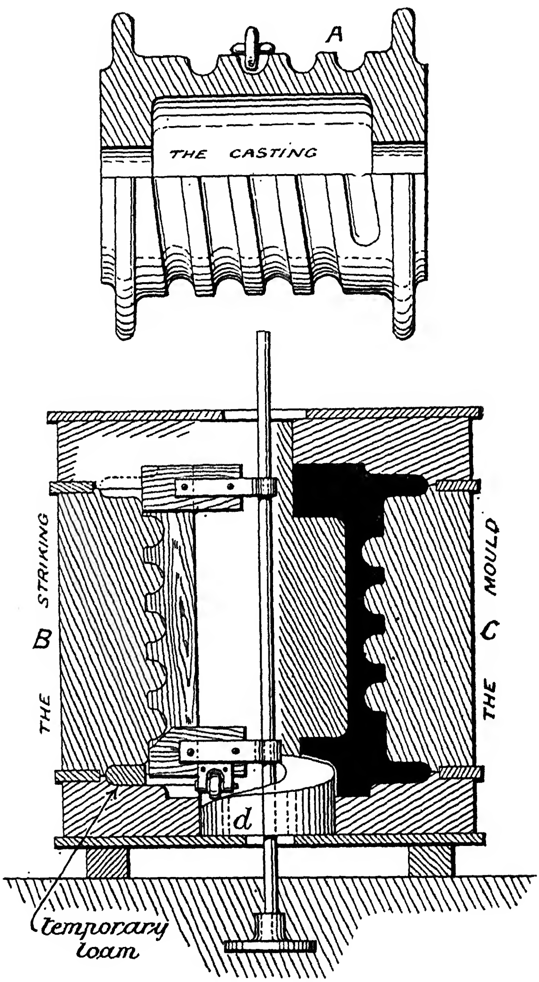


Fig. 73c.

Chain Barrel in Loam.

The rest of the mould is self-explanatory with what has gone before, and is entirely formed by loam boards.

Rapping plates have become a necessity in order to prevent injury to the pattern by the moulder. They are shewn at Fig. 73*d*, being let into the pattern, and are screwed to receive a lifting rod as there shown.

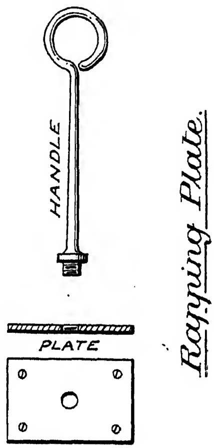


Fig. 73d.

✓ **Crystallization** of cast iron.—During the cooling of a casting the crystals arrange themselves in lines perpendicular to the surface, but the interior portion, being cooled more slowly, preserves its granular nature. Fig. 74 will shew the appearance of a bar of cast iron when broken longitudinally (the student should clearly understand that the markings are exaggerated).

If the corners of the casting are made quite sharp the crystals will be abruptly turned at these places, and, meeting each other also abruptly for some distance below the surface, namely, as far down as they are formed, will create a *line of fracture* or portion weaker than the rest. Whether these corners be external or internal, matters not; the same thing happens. Fig. 75 shows other examples having 're-entrant angles,' as they are called, A being a circular boss cast on a plate, and B a cylinder with flanges. It will be clear that breakage would always occur more easily at these sharp angles.

When the Menai Bridge was built, the hydraulic press made for the purpose of lifting the 'tubes' had a flat bottom with pretty sharp corners, as will be understood from Fig. 76, which is a sketch of the press first used. Stephenson took the precaution of building up at each 10 inches of lift, and, had it not been for this, great damage might have occurred, for the bottom of the press suddenly gave way, and the tube fell through a space of ten inches. Fig. 77 represents the press since adopted, the crystals being allowed to take a gradual turn, so as to leave no line of fracture.

It is a general law that there should be no abrupt corners in a casting, either external or internal, principally for the reasons

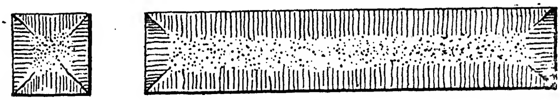


Fig. 74.

RE-ENTRANT ANGLES.

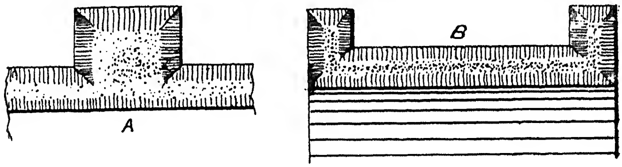


Fig. 75.

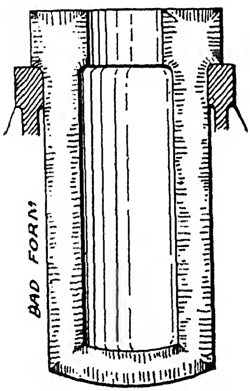


Fig. 76.

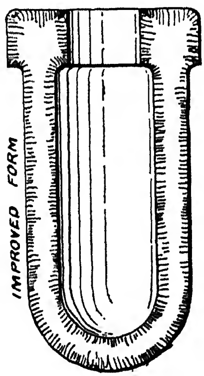


Fig. 77.

Crystallization.

already given, and also to permit of an easy flow of the metal, and prevent the breakage of corners of sand.

Warping and Shrinkage of Castings.—The general effects produced by unequal shrinkage during cooling should be well understood in designing a casting. These may be pretty well arrived at by the consideration that, other things being equal, those places will cool last where the largest amount of metal is aggregated. Our first rule, therefore, is to endeavour to keep the casting uniform in thickness. For unequal cooling is sure to produce internal strains, and *that portion cooling first will set, and be compressed by the contraction of the part that is still cooling.* Besides, if a thin part join a thick part very abruptly, the cooling may produce such strains as to break the thin piece away altogether. We ought therefore to make the juncture of unequal thicknesses as *gradual* as possible.

Take a **Plate**, Fig. 78, lying on a surface of sand. The top part cools first, on account of being open to the air, while the under surface is still in contact with the hot sand, and the effect of cooling is to make the plate convex on the upper surface, by the after contraction of the lower surface.

In a **Hollow Cylinder**, Fig. 79, the heat cannot pass through the core so quickly as it can from the outside, so the latter cools first, and the cylinder is made barrel-shaped by the contraction of the interior. We must also note that the outer layer will be in compression (see Fig. 83), which is a cross section.

A **Solid Ball** will be found porous on the inside, if broken, because the shell sets first, and the internal metal, being thus held fast, is bound to leave vacuities on shrinking.

A **Girder** of the form sketched in Fig. 80 will curve longitudinally in cooling, for here the most metal is collected in the larger flange, and the casting is therefore pulled together on that side, after the top web has cooled.

A **Pulley** with a thin rim, as in Fig. 81, will cool last at the centre boss, and so produce a compressive strain in the rim; if therefore a piece were broken out at A it could not be returned.

Shrinkage occurs while the metal is cooling from a red heat downward, and the moulder can do a great deal to prevent it occurring unequally by uncovering at the red-hot stage those portions of the casting which are likely to retain heat longest, and by keeping others covered, for equal cooling means equal shrinking.

Fig. 78.

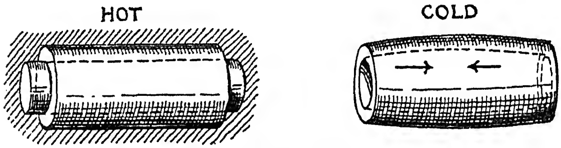


Fig. 79.

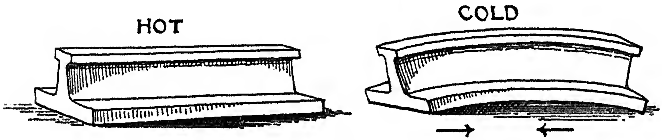


Fig. 80.

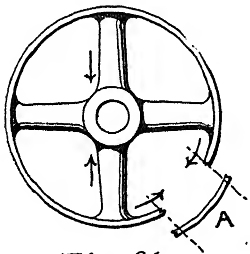


Fig. 81.

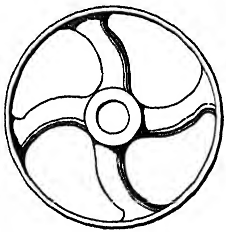
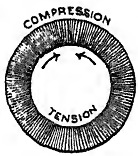


Fig. 82.

Fig. 83



Unequal Shrinkage in castings.

Hollow cylinders of all kinds are better moulded by inserting an iron tube in the core, through which cold water is allowed to circulate, and this can be so regulated as to produce a tensile strain on the outside metal if needed, or, what is better, no strain at all.

The pulley previously mentioned can be improved by curving the arms, as in Fig. 82, thus giving them sufficient elasticity to take the strain off the rim ; and such an example as the girder must be left to the moulder's ingenuity, the thicker portions being first uncovered, so as to cool quickly.

Of course it must be understood that a casting is never broken directly by compressive stresses, but only by tensile stresses induced by them in some other portion of the casting. Thus the pulley will break at the arms, in which a tension is induced, rather than at the compressed rim, although the latter may be thin. This observation is true for all cases. (*See p.* 1012.)

(For Plaster Patterns see Appendix II., p. 781.)

CHAPTER III.

METALLURGY AND PROPERTIES OF MATERIALS.

It will be well, so as to avoid repetition in succeeding chapters, to digress somewhat, in order to consider the properties, and to some extent the metallurgy of the materials used in mechanical engineering, omitting only the consideration of their strength, which will be treated of in the second part of this work.

These materials may be classified as follows:—

1. Cast Iron.
2. Wrought Iron.
3. Cast Steel.
4. Forged Steel.
5. Copper.
6. Zinc.
7. Tin.
8. Gun Metal.
9. Brass.
10. Phosphor Bronze.
11. Muntz Metal.
12. Manganese Bronze.
13. White Metal.
14. Wood.*

But we must first become acquainted with such chemical elements as are necessary to understand the processes we intend to consider. Such are: Carbon (C), Silicon (Si), Iron (Fe), Sulphur (S), Phosphorus (P), Manganese (Mn), and Oxygen (O).

Carbon is an *allotropic* element, that is, it exists under different forms, which are: Charcoal, blacklead, and diamond. The first is pure carbon, and so is coke, or nearly so. The second is not *lead*, and is also called *graphite* and *plumbago*; and the third is the crystalline form. If carbon is allowed to unite with oxygen it forms carbon dioxide (CO_2), a gas. Carbonic oxide, or Carbon monoxide (CO), is another gaseous combination of carbon and oxygen.

A chemical *combination* is the union of elements in such a way that they can only be separated by chemical action, while a mechanical *mixture* requires only mechanical means (very often filtration) to break it up.

* For further materials see Appendix II., pp. 794–801.

Silicon exists in combination with oxygen as silicon dioxide or *silica* ($Si\,O_2$), and is so found in the crystals of sea-sand; glass is a mixture of several silicates. It is of value in cast iron.

Iron is found in combination with oxygen, the ore being termed a ferric oxide ($Fe_2\,O_3$), but it may be rendered quite pure by chemical and mechanical means.

Sulphur is well known in the form of brimstone, and is an impurity in iron, producing red-shortness.

Phosphorus is also an impurity, producing cold-shortness, while **Manganese** is of value only when mixed with iron and other metals in certain definite proportions.

(1.) **Cast Iron.**—There are seven varieties of iron ore, containing from fifty to seventy per cent. of iron in their composition. The blast-furnace (Fig. 84) is used for smelting the ore, which is

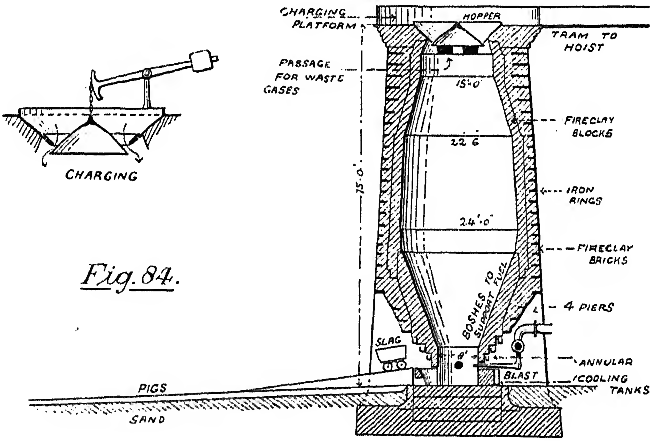


Fig. 84.

A Tees-Side Blast Furnace

done at a very high temperature, with coke as fuel, and lime or clay as a flux.* The molten iron is run into pigs, while the slag

* Lime is the usual flux, but clay is sometimes required, as in the case of hematite ore, and then is applied in the form of clay ironstone.

formed by the combination of the flux with the impurities of the ore, is separately withdrawn. (*See Appendix II., p.* 788–9.)

The action in the blast-furnace is this:—Air being introduced by the blast to give us oxygen, and coke to provide carbon; then, if the coke be heated to redness, *carbon dioxide* is formed,

$$\underset{\text{From Air.}}{2O} + \underset{\text{Coke.}}{C} = \underset{\text{Gas.}}{CO_2}$$

As this gas ascends it takes up carbon from the coke, which it passes on its way, thus:

$$\underset{\text{Carbon diox.}}{CO_2} + \underset{\text{Coke.}}{C} = \underset{\text{Carbon monox.}}{2CO}$$

And we now have *carbon monoxide.*

Ascending further, this last-mentioned gas meets the iron ore, which is now at a great heat. The oxygen in the ore has then the choice of remaining where it is (Fe_2O_3) or of combining with the CO; preferring the CO, it forms with it *carbon dioxide* once more,

$$\underset{\text{Carbon monox.}}{3CO} + \underset{\text{Ore.}}{Fe_2O_3} = \underset{\text{Carbon diox.}}{3CO_2} + \underset{\text{Iron.}}{2Fe}$$

And the iron is now left, but in a viscous condition. As it takes up carbon, however, it becomes more fluid, descends to the bottom of the furnace, and may be then run out.

Other substances have also been absorbed, which may be seen on reference to the table at the commencement of Chapter I., shewing the general composition of the different pigs—grey, mottled, and white.

Sulphur produces red-shortness in cast iron, that is, makes it brittle when red hot, and *Silicon* and *Phosphorus* cold-shortness, or brittleness when cold.

Carbon increases fluidity at the expense of strength, and *Manganese* seems to have the effect during the smelting of inducing the *combination* of the carbon with the iron, thus tending to prevent the formation of graphite.

(2.) **Wrought Iron.**—The white pigs are broken up and subjected to the processes of *refining* and *puddling.* As, however,

these are chemically the same, and the preliminary refining is very often dispensed with, we will give our attention simply to the preparation of wrought iron by puddling.

The object of puddling is to eliminate the graphite entirely, and the combined carbon so far as to leave only about ·25 per cent., which actually increases the strength of the iron.

In the rolling mill, where the metal is rolled into plates or bars, scales of oxide of iron ($Fe_2 O_3$) are formed by the contact of the hot iron with the air. These scales, being broken off, are collected for the puddling furnaces, their use being that of absorbing the carbon from the iron, exactly in the way already described for malleable cast iron.

Being intimately mixed with the broken white pig in the puddling furnace, Fig. 85, and subjected to a powerful flame, the O from the oxide unites with the C of the iron, and passes away as CO_2 gas. Any silicon that is present in the iron unites at the same time with some O and forms SiO_2, so that the iron is left comparatively pure. During the process, the iron is in the form of a spongy mass, and absolute contact of it with the scales of oxide, now liquid, is ensured by the introduction of a long *rake* through a small opening in the side door, for the purpose of stirring the whole well together.* As the puddling nears completion, the metal is kneaded by the rake or *paddle*

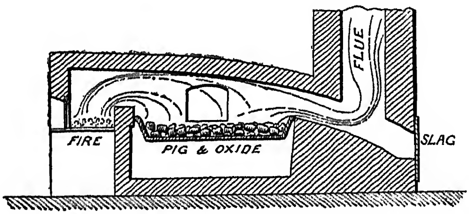


Fig. 85.
Puddling Furnace.

* To avoid hard labour and increase the output, there are many mechanical furnaces now in operation, notably Danks' Rotary Furnace and the Pernot Revolving Hearth.

into balls or *blooms*, and these are then removed and compressed under a steam hammer by rapid blows, so as to squeeze out the slag. The blooms are next rolled out and further squeezed by being passed through the rolls of a rolling mill, giving us iron called *Puddled Bar.*

These bars are now broken up and re-worked by hammering and rolling, more or less, depending on the degree of purity and strength which is required, and we thus have the varieties of wrought iron known as—*common*, *best*, *double best*, and *treble best*, which are used for various ordinary forgings, while *Low Moor* iron is required for the fire-boxes of steam boilers and for more difficult forgings.

The purification of the iron obtained in a puddled bar is shewn by the following table, which may be compared with the table showing the composition of white pig (p. 2) :—

Table showing chemical composition of *Puddled Bar*, in percentages.

Iron	99·31
Combined Carbon	·3
Silicon	·12
Sulphur	·13
Phosphorus	·14
	100·00

Wrought Iron during its conversion from the pig, has lost the capability of being cast into moulds, but has acquired a new nature, becoming *viscous* or sticky, and, as a result, may be worked by the smith, when white or red hot, its formation into different shapes being assisted by the property of *welding*, which as cast iron it did not possess. Repeated rolling gives a fibrous quality, making the iron both stronger and more homogeneous or uniform in texture, and these fibres may be seen on breaking a bar of rolled iron, which then has the appearance shewn at A, Fig. 86, while cast iron or even puddled bar gives a granular fracture (B).

Rolling or hammering iron when cold or nearly so gives it a crystalline structure near the surface, so that T iron is not so strong as bar iron, and plates still weaker. Re-heating and slow cooling tends to remove this source of danger. (*See p.* 1013.)

Generally, then, wrought iron is tough, and more capable of resisting vibration than cast iron, its fibrous character giving it also a distinct advantage in the direction of the fibre, which property may be made use of by judicious crossing in the operations of *piling* and re-heating the iron after puddling.

Fig. 86.
Appearance of fracture.

The best forgings are usually made by the *piling* of wrought iron *scrap*.

(3 and 4.) **Steel** is intermediate in composition between wrought and cast iron, thus:

Cast iron may have	2·2	to	5	per cent.	of carbon.
Steel (for casting)	·3	,,	1·7	,,	,,
Steel (for forging)	·25	,,	1·5	,,	,,
Wrought iron	·0	,,	·25	,,	,,

It will be clear, however, that the exact limits between which we may call the substance 'steel,' without intruding on either wrought or cast iron, are very difficult to define, so that we may have steel which is almost as brittle as cast iron, or we may have it on the other hand nearly as soft as wrought iron.

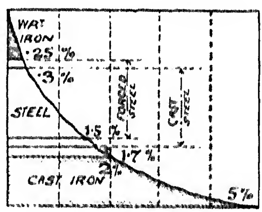


Fig. 87.
Carbon per cent.

Although steel has an intermediate composition, it has not, as we might expect, an intermediate tenacity or use, but is stronger even than wrought iron and consequently more useful. It never has quite the toughness of the best wrought iron, though approaching it closely with mechanical treatment: it is always more homogeneous.

It will also be readily seen that, as steel is intermediate in

composition, it may be made either from wrought or cast iron. We shall first consider the former method.

Cementation.—In this, the oldest process, bars of wrought iron are placed in fire-clay boxes, Fig. 88, with charcoal dust around and between them, and a layer of fire-clay over all (being the *cement* giving the name to the process). They are then subjected to a bright-red heat, for a time varying with the amount of carbon required to be introduced, and which may be as much as a fortnight for the more highly carbonised steels. The charcoal then becomes combined with the iron, and the steel so produced is called *blister steel*, from the fact that the bars are covered with blisters. These bars are next broken up, piled, and heated in a furnace almost exactly like the one in Fig. 85, hammered by rapid blows from a tilt-hammer, Fig. 89, and *shear steel* of a

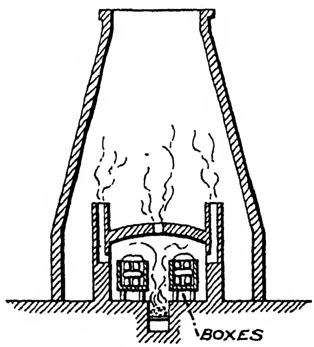


Fig. 88.
Cementation furnace.

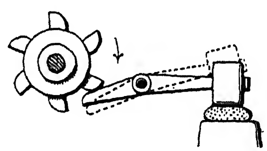


Fig. 89.
Tilt Hammer.

fibrous quality is thus produced.* *Double shear steel* is made by breaking in two and re-hammering *Crucible cast steel* is obtained by melting fragments of blister steel in covered crucibles made of a mixture of fire-clay and plumbago, and placed in sets of six or twelve in furnaces having a similar section to the one shewn in

* The steam hammer is used in later-built works. For drawing, see Chapter IV.

Fig. 50. Several of these crucibles are poured simultaneously to form the ingot, many well-drilled workmen co-operating to do it carefully. This variety of steel is much more homogeneous and has a greater tenacity than shear steel, having a fine granular structure. Brittleness is corrected, and the property of weldability restored by the introduction of manganese in the form of carbonate of manganese.

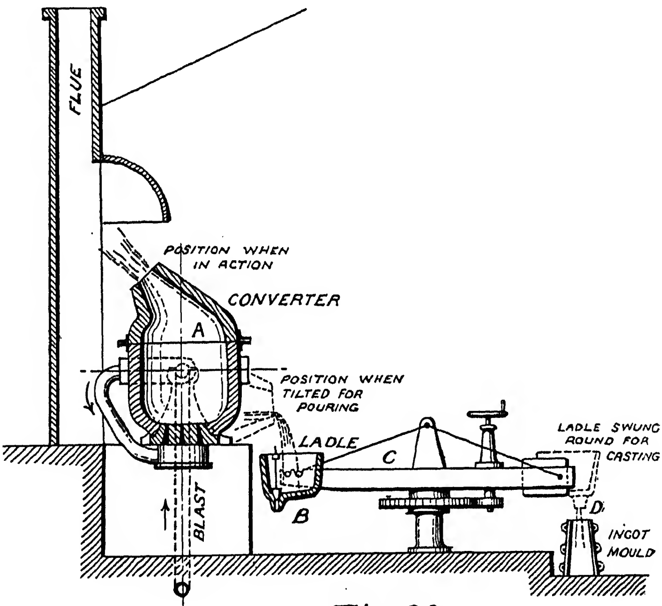


Fig. 90.

Bessemer Steel Plant.

The **Bessemer Process** is used for the purpose of obtaining steel from cast iron. Fig. 90 is intended to shew the necessary plant employed. The *converter* A is filled with molten cast iron, and air is blown through the metal by means of the tuyères at the bottom. The O of the air combines with the C of the iron and passes away as CO_2 gas, leaving the mass as pure iron, the silicon forming a slag (SiO_2) on the surface, which is

separately removed. The temperature must be exceedingly high in order to preserve the iron in its fluid state after the expulsion of the carbon; the entire absence of the latter is discovered by the application of the *spectroscope*, this being the most practical use of that most wonderful instrument. The next operation is the adding of so much carbon as is needed to produce the steel required, and this is done by putting into the converter a measured amount of very pure cast iron called *Spiegeleisen*,* and mixing it well with the metal by re-applying the blast for a short time. The now converted steel is transferred to the ladle B, which is swung round by the crane C, and the metal poured into the ingot through the hole D on releasing the plug at the bottom of the ladle.

The ingots may be afterwards piled and rolled as previously described, to produce a fibrous steel, and if used for forging and welding purposes should not have too much carbon in their composition; or, if required for steel castings, may be re-melted in suitable quantities, much as in the way already mentioned for cast iron.

The Siemens-Martin, or open-hearth process, is carried on in a special kind of furnace, called a *regenerative* furnace, invented by Sir W. Siemens. Fig. 91 is a drawing which will shew all the necessary parts. A is the hearth, sloped in the side elevation, so that the metal may run out when tapped at T. A current of air is allowed to pass under the hearth at C, to prevent the melting of the fire-clay. The combustion of a mixture of common coal gas and air is the source of heat, the arrows showing the passage to the interior of the gas through the valve G, while the air enters through the valve A. In the figure the mixture is seen entering the *right side* of the furnace. Being ignited at J by means of a red-hot bar, gradually and carefully at first, the flame is directed by the roof on to the metal, and the heat passes away by the *left side* of the furnace, returning through the valves and past the damper D to the chimney. Were it not, however, for Siemens' beautiful regenerative principle a great deal of heat would be wasted. The regenerators are shewn at R R *r r*; they are hard fire-clay or silica bricks piled as a grating. The rejected heat from the hearth is intercepted by those marked R R, so that

* See Appendix II., p. 790.

although the mixture enters at 700° F. the products of combustion pass to the chimney at 200° or 300° F. In a short time the bricks become white hot, and the valves A G are then reversed as is shown at V_1 and V_2, the former being the position for the action already mentioned, and the latter allowing the gas and air to

Siemens' Regenerative Furnace.

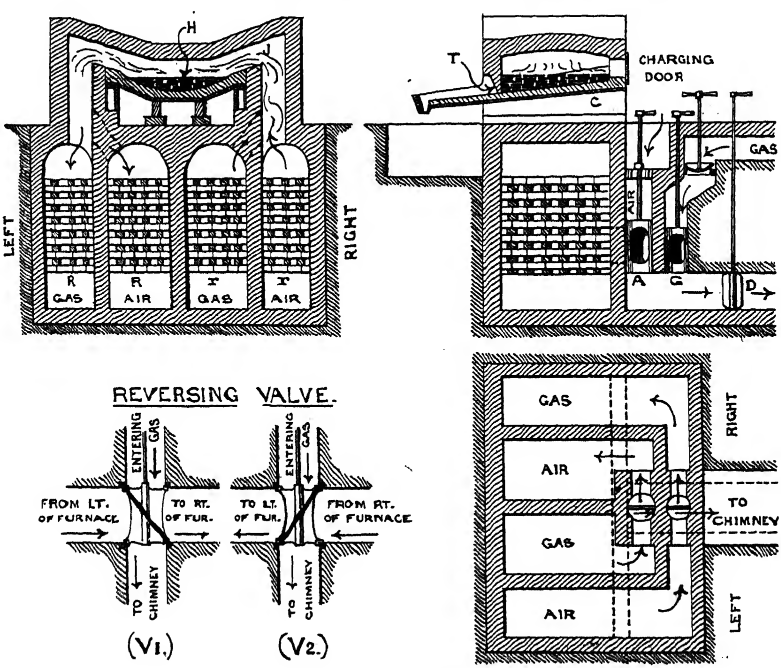


Fig. 91.

enter first the *left side* of the furnace and leave on the opposite side. Doing so, it is evident that the heat which was absorbed in the last operation by the regenerators R R is now taken up again by the entering gases, and the bricks *r r*, in their turn receive the rejected heat.

By this means a large amount of heat is made useful which

would otherwise be wasted, the valves being reversed regulɛ whenever the bricks acquire too much heat.

The furnace is first charged with pig iron, and when thi: melted, heated wrought iron and steel scraps are added degrees (these three in nearly equal proportions). When all thoroughly mixed a little piece of cast iron, in the form spiegeleisen, is added, together with a very little manganese.

Experience, the principal guide for this mixture, is again call into play immediately before completion of the operation, t foreman trying small samples taken from the furnace and cool in water, by breaking them and examining the fracture. If sat factory, the steel is now poured into ladles by tapping at T.

As soon as the metal ceases to flow easily it is known th there is only slag left. The ladle is then removed, and the sl allowed to run to the ground or into moulds.

The **Landore-Siemens** process, also the patent of Sir Siemens, differs in the fact that iron ore is used direct. On bei first reduced and the slag got rid of, it forms spongy balls malleable iron, which are then dissolved in molten pig iro spiegeleisen being added as before. It often receives the nan of the 'pig and ore' process.

In the **Siemens** process the ore and flux are mixed dire with the pig; more slag is therefore produced.

Steel Castings made by any of the above methods mu be annealed slowly in a closed furnace for a week or mor to prevent cold-shortness. *Honeycombing*, or the presence vacuous spaces in the metal, is the principal trouble, and is partl prevented by the addition of silicon, as silico-ferromanganes but is only perfectly got rid of by the Whitworth process, wher the molten ingot is compressed by powerful hydraulic pressur until it is quite set. The great advantage of this compressior which amounts to from six to twenty tons per square inch, i shewn by the fact that the ingot is made to contract as mucl as one-and-a-half inches per foot of length. The mould consist of a steel cylinder, lined with refractory material, and so con structed that when placed under an hydraulic press, the gase may escape through the sides of the mould. (*See App. II., p.* 790.)

We may always expect highly carbonised steel to be deficien

in toughness, and therefore inferior to wrought iron in that respect. It may be improved by the annealing spoken of, but steel that is required for boiler or bridge work must be capable of resisting vibration, and so a milder quality is used, which, though it may not be considerably stronger than iron, is more homogeneous and has a finer grain.

The amount of carbon varies with the use to which the steel is to be put, and is shewn by the following table :—

Razor temper......	$1\frac{1}{2}$ °/₀ Carbon	Will spoil with over-heating.
Sawfile ,, ...	$1\frac{3}{8}$ °/₀ ,, ...	To be heated only cherry red.
Tool ,, ...	$1\frac{1}{4}$ °/₀ ,, ...	May weld with great care.
Spindle ,, ...	$1\frac{1}{8}$ °/₀ ,, ...	Ditto.
Chisel ,, ...	1 °/₀ ,, ...	Tough ; will harden at low heat.
Sett ,, ...	$\frac{7}{8}$ °/₀ ,, ...	Stands hammer ; welds easily.
Die ,, ...	$\frac{3}{4}$ °/₀ ,, ...	Stands pressure ; welds like iron.

Cutting tools require most carbon. (*See Appendix II., p.* 793.)

Tempering, or the capability of receiving any degree of hardness, is a property of steel, and was formerly applied as a test to distinguish it from wrought iron; while *case-hardening* is a method of partially converting wrought iron into steel, but both these subjects will be reserved for our next chapter.

Test.—A rough test to distinguish between wrought iron and steel is to put a drop of dilute nitric acid on the polished metal, when a greenish-grey stain will indicate iron, and a black spot will shew steel; the purer the black, the more carbon may be suspected, so that we may even get a notion of quality.

(5.) **Copper** Ore is various in character, but may have iron, sulphur, antimony, or arsenic associated with it. The operations are three in number :—(1) *Roasting*, to get rid of arsenic and sulphur, the iron forming an oxide. (2) *Smelting*, to dissolve the iron oxide, and leave copper combined with sulphur. (3) *Roasting and Smelting*, to remove the sulphur and obtain metallic copper. The furnaces used throughout are of the same class as the puddling furnace, Fig. 85, and called *reverberatory* on account of the arch *beating back* the flame. Other refining processes have

to be gone through before the metal is considered fit for the market.

The metal thus obtained is rolled into plates and hammered to any shape. Besides its malleability it is exceedingly ductile, being easily drawn into wires; it becomes brittle if hammered cold, but its tenacity may be restored by annealing.

Copper is an expensive material, and is only used for pipes that require bending cold, and for fire-boxes, where ductility as well as power to conduct and resist heat are needed: it must be remembered, however, that copper loses its strength somewhat with increase of temperature.

It is also very useful for electrical purposes, being, next to silver, the best metallic conductor. (*See Appendix II.*, *p.* 793.)

(6 and 7.) **Zinc** and **Tin** are of little importance singly to the mechanical engineer.

(8.) **Gun Metal** is an alloy of *Copper and Tin,* and is often called bronze. The proportions are varied for different purposes. Thus to make 100 parts:—

	Copper.	Tin.	
Soft gun metal requires	90	10	(General Ordnance purposes.)
Hard gun metal ,,	82	18	
Bell metal ,,	80	20	

Usually some zinc is added to make the metal more malleable, as:—

Copper	84·22
Tin	10·52
Zinc	5·26
	100·00

Gun metal produces fine castings, and being much stronger than cast iron, is almost the only other metal preferred besides cast steel, for the castings required in modern gunnery. It is often in its harder form made into bearings for shafts. Both strength and toughness are increased by rapid cooling.

(9.) **Brass** is made by alloying *copper* with *zinc.* The pro-

portions vary somewhat, depending on the colour and strength required.

	Parts Copper °/₀	Parts Zinc °/₀
Fine yellow brass has ...	66·6 ...	33·3

The proportion of copper may vary from 66 to 70 per cent., or even higher. A little lead is sometimes added. Brass is principally used on account of its fine colour, and because it is easily tooled.

(10.) **Muntz Metal** is a brass having the proportion of 60 per cent. of copper and 40 per cent. of zinc. It is largely used for bolts in marine work that are liable to rust, and especially for pins that have to turn in their sockets, on account of its great strength, as well as the faculty of being forged, which it possesses.

(11.) **Phosphor Bronze** is, like gun metal, a mixture of copper and tin, but with the addition of a small measured quantity of *phosphorus*. (*See Appendix I., p.* 748.)

Its strength is then so much increased as to be equal to that of wrought iron, and it has consequently been extensively used, within recent years, where strength is required, coupled with intricate form, such as must be cast rather than forged; as for example, toothed wheels subjected to shock. Gun metal is deteriorated by subsequent meltings, while phosphor bronze may be re-melted without injury.

It has considerable ductility, and may be formed into wire, and used for spiral springs subjected to steam or water.

(12.) **Manganese Bronze,** introduced later, is really a yellow brass, to which about 7 lbs. of cupro- or ferro-manganese have been added per cwt. of the metal. The strength is thereby still more increased; and it is used now for a variety of purposes where strength and ductility are required combined, such as hydraulic pipes, which may be then drawn considerably thinner than copper ones; and it is advantageous in many other cases, as may be understood from the fact that it may be both cast into moulds and forged under the hammer. It can also be used to resist rust, so as to keep nuts and bolts free that would otherwise seize. (*See App. II., p.* 801.)

(13.) **White Metal,** otherwise white brass, and in America Babbitt's Metal, or 'Babbit,' is an alloy used for lining bearings. Tin is the principal metal used, and is mixed with copper and

antimony in varying proportions, the following percentages being principally used :—

Copper	8	3
Tin	84	90
Antimony	8	7
	100	100

One advantage of white metal for bearings is that it can be run into the bracket when the journal is in place, and so ensure a good fit. It causes considerably less friction than brass or bronze.

To sum up then, alloys of copper and tin are termed *bronzes*, and may have a little zinc added up to about 1½ or 3 per cent. Those of copper and zinc are called *brasses*, Muntz metal being one of them; and those having tin and antimony, with a little copper, are *white metals.*

Brazing.—Brass or gun metal may be united by this process, which is also termed *hard soldering;* and the joint will then be as strong as the original casting.

Iron or steel may be also connected by brazing if more convenient, especially finished pieces of work. The method is to first carefully clean the work with acid, then take some brass filings and mix with powdered borax as a flux, the borax being preferably moistened with water.* The filings are placed between the parts to be brazed so as to form a joint, as much surface being used for the latter as possible, and the two are held together in red-hot tongs, having thick jaws to keep the heat. The tongs will melt the filings and grip the pieces until perfectly set, and the whole may be finished off in the vice.

If the work cannot be easily gripped, another way is to insert the filings as before, and, binding with iron wire, place the pieces in a clean coke fire until the operation is complete.

Or, still another method is to use the blow-pipe. Here a fine tongue of very hot flame is directed on to the work by blowing with this instrument through a lighted 'Bunsen.'

* More generally, instead of brass filings, 'spelter' is used, which is a powdered brass, containing about equal parts of copper and zinc, and specially manufactured for brazing purposes.

(14.) **Wood** is not used to so great an extent as formerly. Roofs are made of wrought iron; and men-of-war of iron and steel instead of oak: pillars of cast iron: while morticed wheel teeth are almost out of fashion. Brake blocks, too, are made of cast iron, to give a longer time of wear; and wooden buffer beams for locomotives are now being discarded.

Little then need be said of wood. For pattern-making, as already stated, *pine*, *mahogany*, *cherry*, *sycamore*, *lime tree*, and *walnut* are the woods used. *English oak* is the best for beams, but American oak is much cheaper, and the latter is used for the framing of railway and traction waggons, and for locomotive buffer beams. *Ash* is also much employed in waggon work, especially for cart shafts. Mortice teeth are made of *beech* or *hornbeam*. *Lignum vitæ* is of great service for bearings that are immersed in water as, for example, with the screw-propeller and some turbines. (*See Appendix II.*, *p.* 787.)

Railway sleepers are rendered very durable by impregnation with *creosote* or black oil, air being first sucked from the pores of the wood. The creosote is then forced in at great pressure.

The following table gives the melting points in degrees Fahrenheit of the principal metals mentioned in this chapter:—

Cast Iron	2100°F.	Zinc	773°F.
Wrought Iron......	3000° *	Tin.................	442°
Steel	2700°	Gun metal	1900°
Copper	1996°	Brass......	1700° to 1900°

Soft Soldering.—*See Appendix V.*, *p.* 970.

* Castings of 'wrought iron' have been made, though the process is somewhat intricate, and has not been extensively applied. The method consists principally in lowering the high melting point of wrought iron by the addition of aluminium. Swedish wrought iron is used, and from $\frac{1}{2000}$ to $\frac{1}{700}$ of its weight of aluminium is mixed with it.

CHAPTER IV.

SMITHING AND FORGING.

WROUGHT iron is formed into the required shape by *drawing down* and bending while hot; but if there should be insufficient 'stuff,' or if it should be more difficult to entirely finish by drawing down, recourse is had to *welding*.

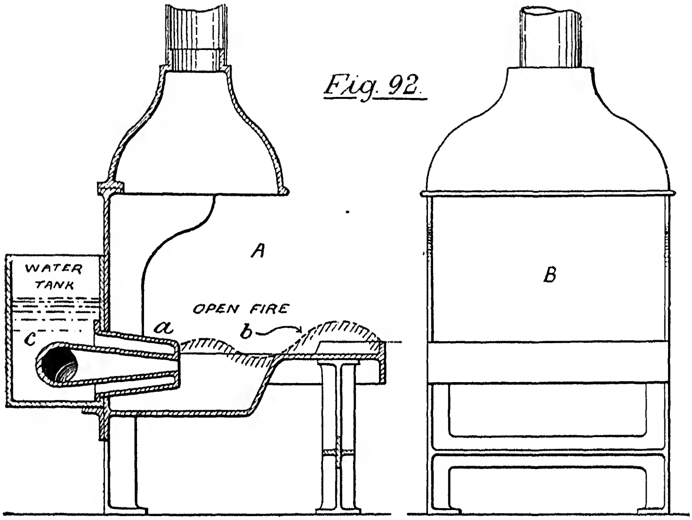
The working of de-carbonised iron may be best treated under two heads, *smithing* and *forging*. The first includes the making of such smaller objects as can be conveniently done at a smith's fire, while the second term may be applied to the shaping of all articles that require heating in a close furnace, and finishing under a heavy steam hammer. In either case the result is denominated a *forging*.

THE SMITHY.

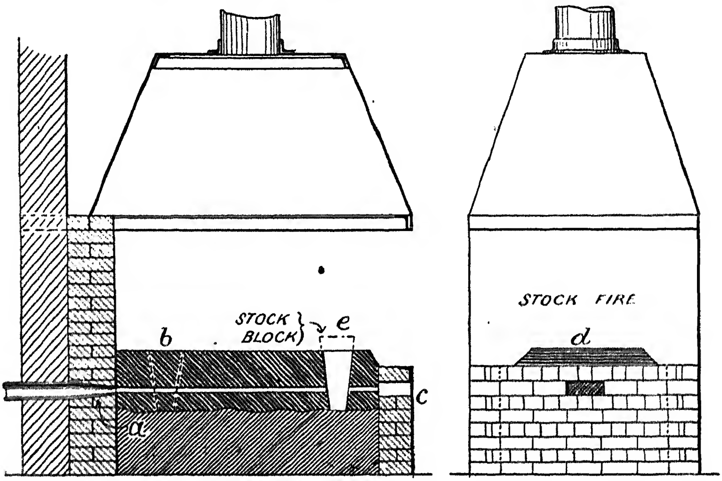
We will first consider shortly the plant and tools employed by the smith.

The Hearth.—This is represented in Fig. 92. A is a sectional elevation, and B a front view. It is necessary to explain here that the smith may arrange his coal on the hearth in two distinct ways, the one being called an 'open' fire, and the other a 'stock' fire. The hearth shown in Fig. 92 is by Messrs. Handyside, and is of iron throughout. It is only adapted for 'open' fire working, being short in length from *a* to *b*. *a* is the tuyère or blast nozzle, constantly surrounded by water contained in the tank *c*, so as to avoid burning at the outlet, or the accumulation of caked slag. The work to be heated is placed in the hollow portion of the hearth surrounded by coal, and as the coal burns away more is supplied from the hillock *b*. It will then be seen that there is no special difficulty in arranging the coal for 'open' fire working. 'Stock' working requires a certain amount of trouble in first preparing the coal, which is usually done first thing every morning. After this first preparation it will, however, keep in working order

Fig. 92.



Cast-iron Smiths' Hearth. (BY AND. HANDYSIDE & Co)



Brick-built Smiths' Hearth.

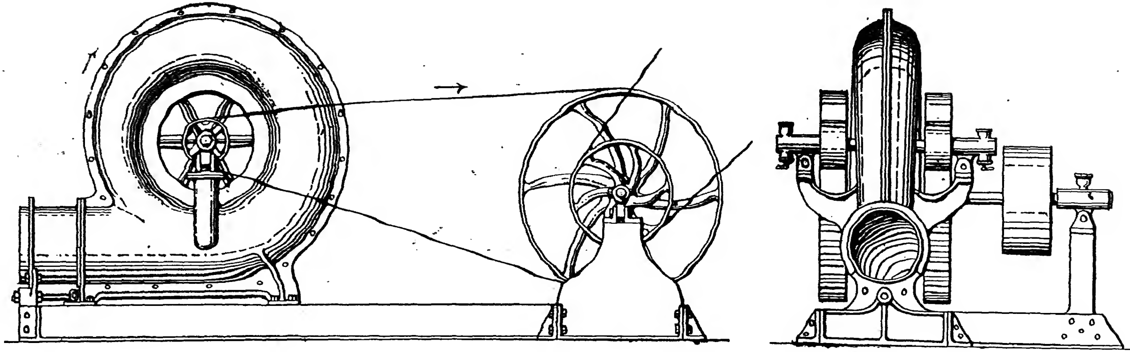
Fig. 95.

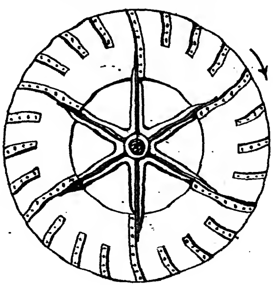
for the rest of the day, and has many advantages, as will be seen Fig. 95 represents an ordinary smith's hearth, built up partly of brick and partly of iron. *a* is the blast nozzle, which need not now be surrounded by water, because the fire will never be nearer to it than the position marked *b*, and so no caking can happen. In building the 'stock' a loose brick is first taken out at *c*, and a bar passed through and inserted in the tuyère. The coal is now damped by sprinkling water upon it with a wisp of straw, and is built up into the form shown, the ridge *d* being neatly flatted down, by using the back of the shovel. Beginning at the tuyère and advancing frontwards the 'stock' is finished round the piece of wood *e*, which is called the 'stock block.' We may now remove both bar and block, and make the fire in the space *e*. The iron to be heated is placed in this space and covered up with loose coal, which is always brought from the front end *c*, so that the stock gradually burns away to the end *b* by the close of the day. The advantages of 'stock' working are these: (1) we need no water tuyère nor consequent attention to water supply; (2) the bar to be heated is only acted on by the fire to the length required (whereas 'open' working has a tendency to heat it to a greater extent); and the method is generally more economical.

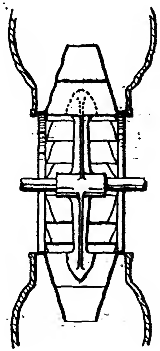
The Blast.—Air is constantly supplied to the fire, when working, by means either of bellows, fan, or blower, one of the latter two being in use at an engineers' smithy, where all the fires are connected to one main blast pipe. Fig. 95A, Plate I., represents a fully equipped smithy, as designed by Messrs. Handyside, and fitted with their hearths throughout. The main blast pipe is shewn by the dotted lines in plan.

Fig. 93 is a drawing of a **Fan** by Sturtevant. There should be a good large space left beyond the vanes, to allow the velocity energy given to the air by them to be easily transformed into pressure energy in the pipe, and so prevent waste by eddies.

Roots' Blower, as made by Messrs. Samuelson, of Banbury, is shewn at Fig. 94. The air in this machine is literally scraped out of the casing on the side A, by the revolution of the two figures S S, in opposite directions, and is delivered at B, a fresh supply replacing the partial vacuum formed. The rollers, as the above figures are called, are compelled to work







Fan for Smithy
or Cupola.

(BY STURTEVANT)

Fig. 93.

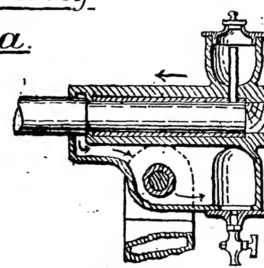
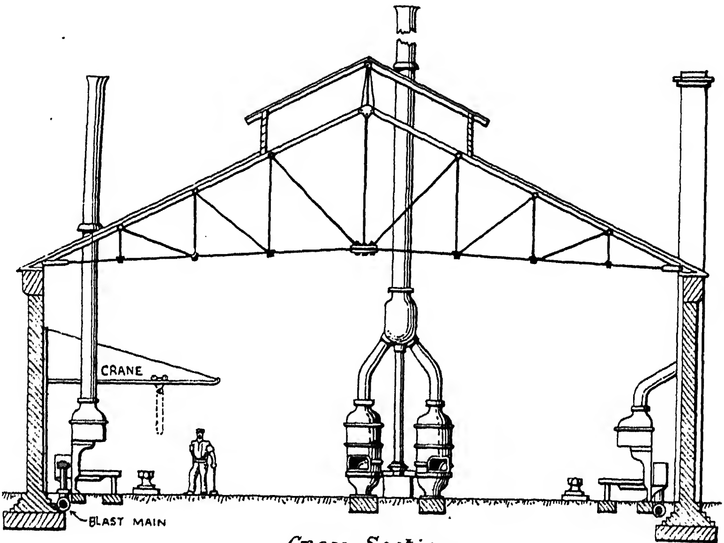


PLATE I.

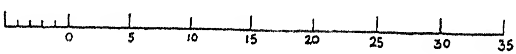


Cross Section.

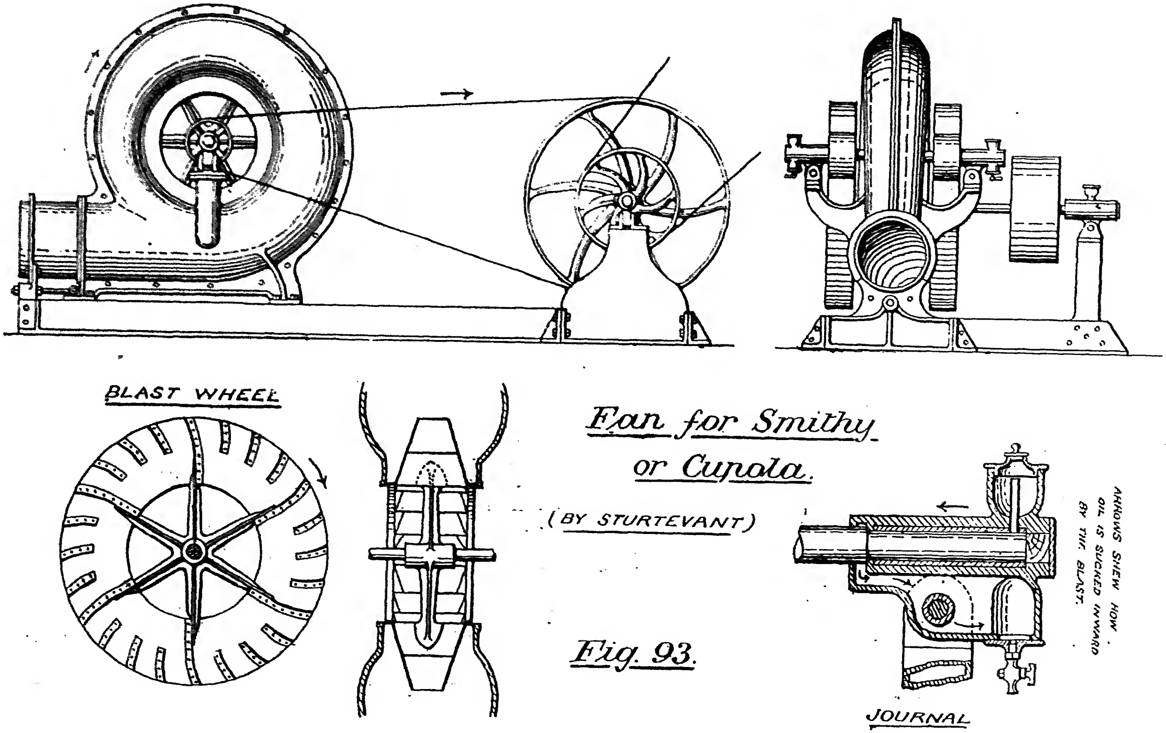
ARRANGEMENT OF AN ENGINEER'S SMITHY

(fitted with Handyside's Hearths.)

Fig. 95a.



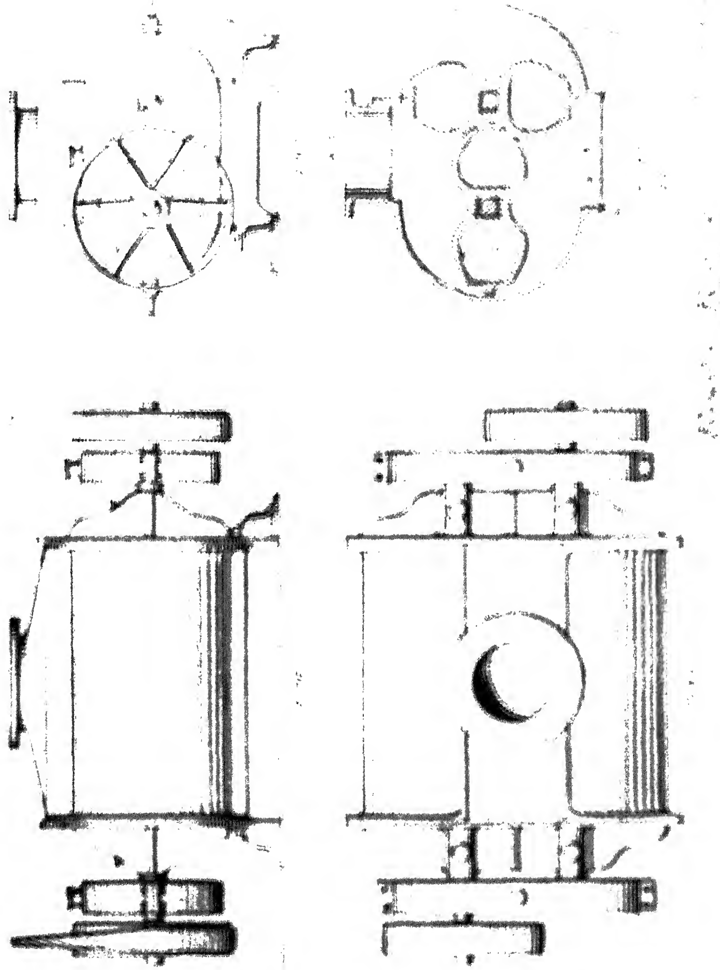
SCALE OF FEET.



Fan for Smithy or Cupola.

(BY STURTEVANT)

Fig. 93.



correctly by the introduction of equal spur wheels, shown [illegible] at c c, being covered by a casing, and the shafts are rotated in opposite directions by means of crossed and open belts, as shown

in the side elevation. The power absorbed in running this machine is very slight, and the speed need not be more than 300 revolutions per minute. A fan, on the other hand, to be effective, must be driven at a great velocity, say from 1000 to 2000 revolutions per minute; more shafting and pullies are required, as shewn in Fig. 93, and the percentage of loss by friction is consequently high. The blower is, however, very noisy.

✓ **Tools.**—Among these we must first mention the **Anvil**, Fig. 96. It is made of wrought iron, and has a surface of steel about a quarter of an inch thick welded on at A to form the top face; B is the *beak* or *horn;* C and D are square holes to receive 'bottom' tools, and E E are used in punching. At Fig. 96B is illustrated a French anvil. It is not provided with any holes, the swage block (described later) serving instead.

Two kinds of **Hammer** are required: the *hand* hammer weighing two-and-a-half to three pounds, for the smith; and the *sledge* hammer, used by his helper, weighing from eight to fourteen pounds, and even more. If the sledge is only worked by lifting over the shoulder, a short handle is used, say three feet long, but, when swung, in making heavy forgings, a long shaft is required, the right hand being drawn inward to the end as the hammer approaches the work, thus giving the latter the full effect of the stored energy.

Other tools, shewn in Fig. 97, are principally for the purpose of finishing work for which the hammer alone would be insufficient. They often go in pairs, as *top* and *bottom* tools, the smith holding the first by means of a hazel rod wrapped round it, while the second is placed upright in one of the square holes in the anvil. A A are *chisels*, B B *fullers*, C is a *flat-face or flatter*, D a *punch*, and E E are *swages*. The last term is applied to any specially-shaped top and bottom tools designed for the purpose of finishing work with greater ease and accuracy to a particular form, such as round, hexagonal, &c. F is a *set hammer*, having either a square or circular face; it is held steadily on the work while being struck, so that in one sense it is not a hammer at all. It is convenient as a top tool to reduce work or 'set' it down, the anvil serving as bottom tool. G is a '*heading*' tool, useful in making the heads of bolts and pins. It is held by the hand at one end

English Anvil

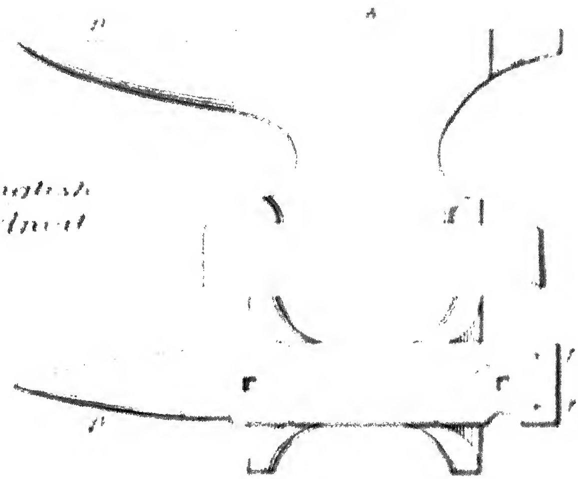


Fig 96a

French Anvil

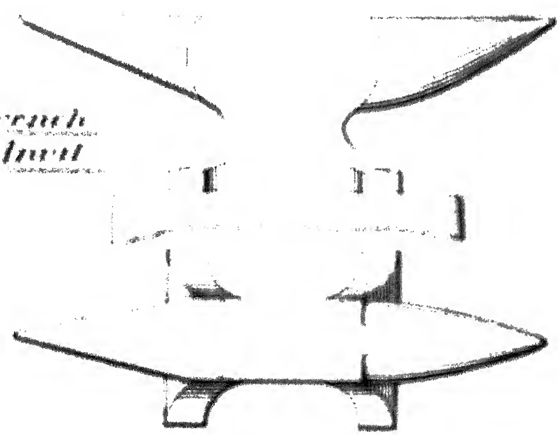


Fig 96b

[illegible] and, being [illegible] the bolt, the head is [illegible] as shown at a, c and b, [illegible] of small levers [illegible] that a good set of tools will [illegible] Several of each of those above mentioned [illegible]

Tongs are [illegible], and have several different forms [illegible] where the work is gripped, such as *flat mouth* [illegible], *round mouth* [illegible], or *angular mouth* [illegible]. They [illegible] take work from a quarter of an inch to three inches thick, and all of them are more useful if bent out at [illegible] to admit a bolt head or collar.

An [illegible] adjunct to the anvil is the **Swage Block**, Fig. [illegible]. It can be set up in any position, and serves to finish several different forms of forging, the holes being for heading purposes. The swage block is usually made of cast iron, though cast steel is now often preferred.

The smith having to hold the work in the tongs with one hand, he wields the hand hammer with the other, where plain forging is required, but, when top and bottom tools have to be applied, he is fully occupied with the top tool and the work to be done, so the hammering must be performed by a helper or *striker*. A good deal of forging may be made by this method, which is called "*working double handed*," especially by using a heavy sledge.

The striker is sometimes dispensed with by introducing some power appliance, and often very ingenious arrangement to take his place. The method followed would then be called "*working single handed*."

Fig. [illegible] shows one of these tools, termed a *dolly*, and its purpose in the figure is round swaging, though other forms could be applied. It is fixed in one of the square holes of the anvil, and is struck at a with the hammer in the right hand, while the left hand holds the work at n with the tongs. (*See App. V., p.* [illegible].)

Bolts or rivets may be forged single handed by employing the block a, Fig. [illegible]. A stock piece, a, is put in the hole in the block so as to fix the length of the rivet b. Pieces of round iron

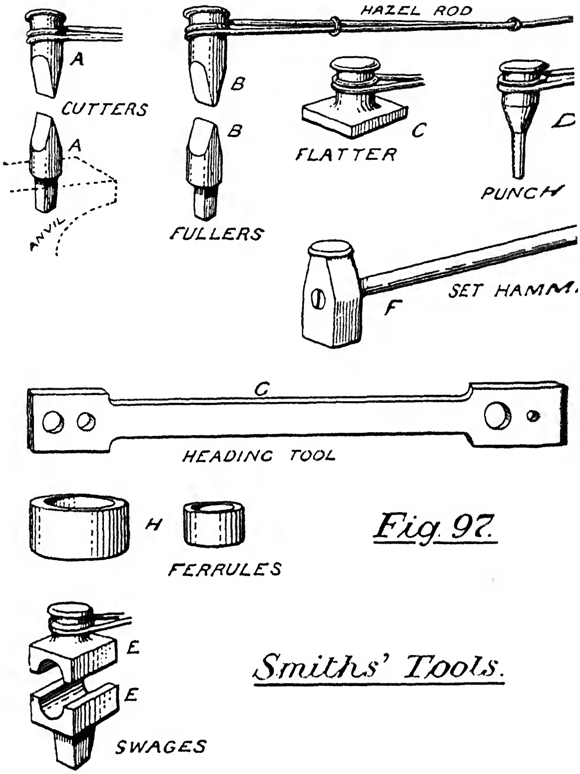


Fig. 97.

Smiths' Tools.

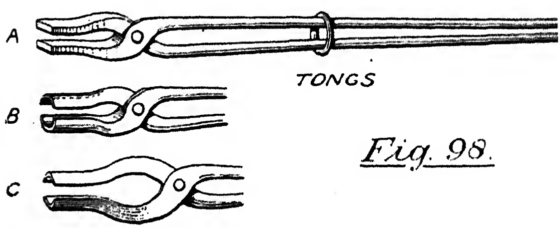


Fig. 98.

are cut off of proper length to form the rivet, and being heated, are dropped into the hole at *b*. The hammer *h* is now worked from the footboard *f*, the blow being delivered by pressing the foot downward on the latter, while the return of the hammer is ensured by the elasticity of the sapling of ash *s*, which is bent on each down stroke of the footboard, and in becoming straight again lifts the hammer. The correct form of the rivet head is given by applying the cupping tool *c*, held in the hand. When the rivet is finished it may be released and thrown out by striking the foot sharply on the lever *l*, which thus takes the dotted position, and the rivet can be then picked up and cooled in water.

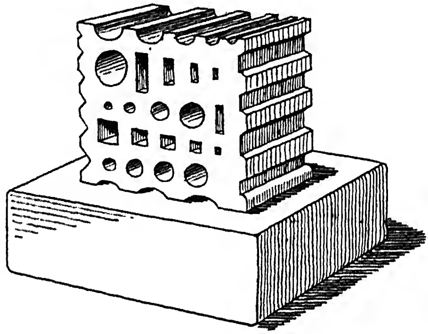


Fig. 99. Swage Block.

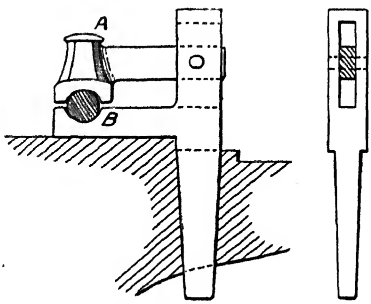


Fig. 100. Dolly

Steam Hammer for Smithy. — Lastly, the smith requires for his heavy forgings the aid of a small steam hammer. We say 'small' to distinguish from the larger type in use by the forgeman, but the smith's hammer is anything but small. The one illustrated in Fig. 101*b*, Plate II., is spoken of as a 10 cwt. steam hammer, and this means that the piston and piston rod A A, the tup B, and the pallett C, together weigh 10 cwts. This, of course, does not take account of the steam pressure, which at 40 lbs. per square inch con-

siderably increases the [illegible] gravity being constant, the [illegible] the anvil may be easily calculated.

Then, if w = weight of the [illegible]
v = velocity [illegible]
g = gravity [illegible]
P = greatest [illegible] pressure [illegible]
p = mean [illegible] pressure [illegible]
d = depth of [illegible] impression [illegible]
$\frac{wv^2}{2g}$ = energy of the [illegible]

And if the depth of the impression [illegible]

$$pd = \text{energy [illegible]}$$

$$\frac{wv^2}{2g} = pd \qquad p = \frac{wv^2}{2gd} \qquad \text{[illegible]}$$

In a perfectly elastic material $P = 2p$, as shown by the diagram of work, Fig. 101a (see *Appendix II*, p. [illegible]).

The pressure may also be found by a consideration of the time during which the energy is given up.

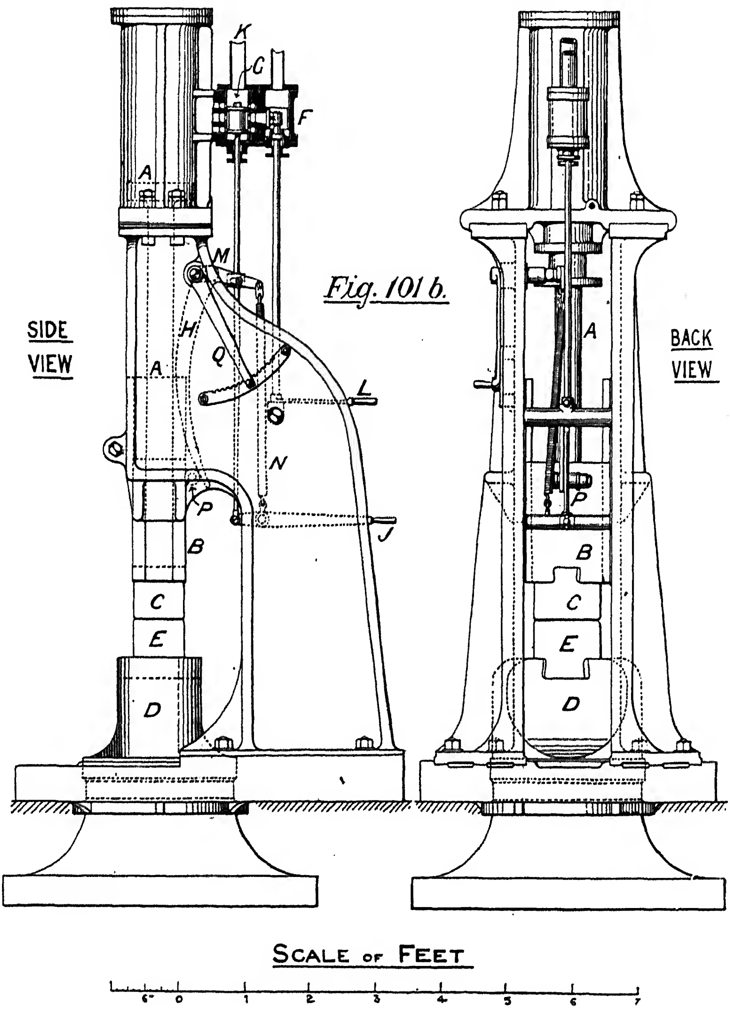
Then (if t = number of seconds of time [illegible]),

$$\frac{wv}{g} = pt \text{ and } p = \frac{wv}{gt} \qquad \text{[illegible]}$$

In practice v, t, and d are difficult to arrive at, both on account of the varying rigidity of the material receiving the blow, and on account of the difficulty of measuring the *elastic* [illegible], or again, of fixing the ratio of p to P, which is only shown in Fig. 101a for elastic deformation; p would have a much larger value in proportion to P in actual practice. Messrs. Massey state, however, that with 90 lbs. steam pressure

A ½ cwt. hammer gives a blow of about 2½ tons.
A 5 " " " " 30 [illegible]
([illegible] p. [illegible])

Referring again to Fig. 101b, B is the anvil and A the anvil pallett. F is the valve for admitting steam, opened by a horizontal movement of the handle I, and G is the regulating valve, which requiring to work easily, is balanced by being made of piston



10 CWT. STEAM HAMMER

(by Messrs. B. & S. Massey)

face p. 98

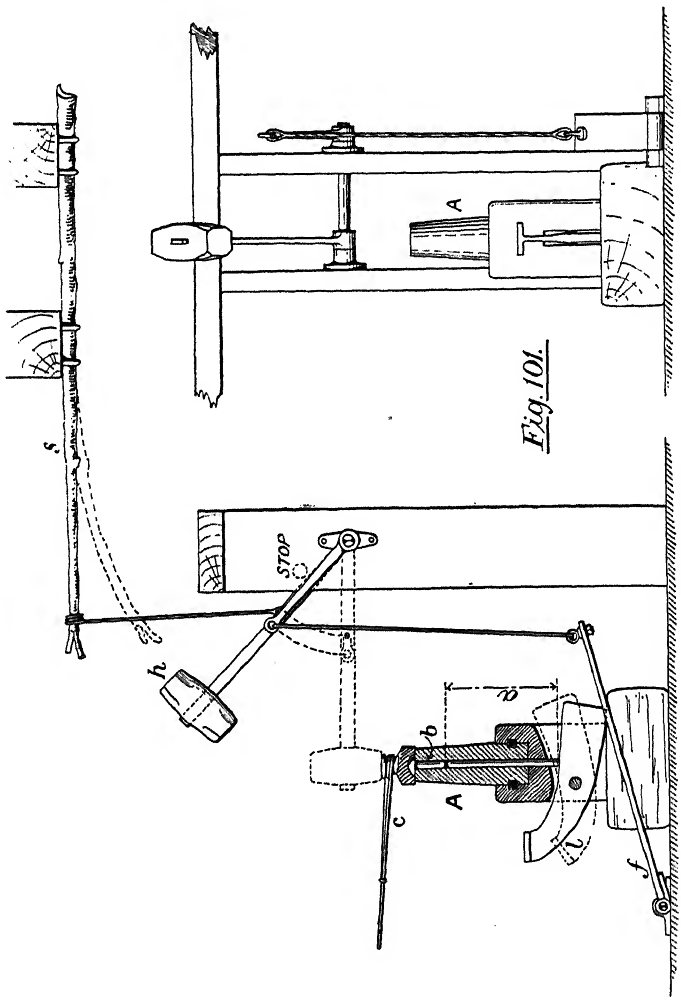


Fig. 101.

Single-handed working.

form. In the drawing, steam is shewn entering by the mid port, and passing down the lower cylinder port to raise the piston; at the same time the exhaust steam from the top of the piston is passing down the upper port and out at K. Now (supposing the self-acting lever H is out of gear), the piston having reached its

[illegible] the valve [illegible] handle, and, when the [illegible] [illegible] [illegible] by [illegible] [illegible]

The [illegible] [illegible] starting valve [illegible] and the [illegible] admit steam at the [illegible] [illegible] valve [illegible]

If the [illegible] there [illegible] sufficiently to the right to be out of the way of the tappet [illegible] The hammer may then be worked by hand. If [illegible] blows are required, [illegible] must be [illegible] to the extreme right, while [illegible] are obtained by [illegible] greater and greater [illegible] extreme left.

If a line of shafting runs along the shop, a hammer like the [illegible] in Fig. [illegible] may be applied with advantage for stamping work, though it is less [illegible] in its beats for regular forging. Here the iron strap [illegible] is made to grip the drum [illegible] by the pull of the [illegible] [illegible], and so the hammer [illegible] is lifted by the power transmitted [illegible] the shaft [illegible]. Directly the handle [illegible] is released [illegible], when the weight reaches the dotted position, the strap [illegible] and the [illegible] is struck. (See *Appendix II*, [illegible].)

Heating. Good wrought iron will sustain [illegible] [illegible] degree of heat without injury, but the more carbonised steels have to be treated with great care, for if heated beyond [illegible] red they begin to break up from red shortness, and cast steel [illegible] overheat very easily. The student may refer to Fig. [illegible] Chapter III., which explains diagrammatically the different amounts of carbon in wrought iron, steel, and cast iron.

In forging wrought iron or steel, the heat should be kept up, and a new heat taken when the work is getting too cold, for it is generally admitted that working any metal by cold hammering crystallises the surface receiving the blow, and if annealing be not

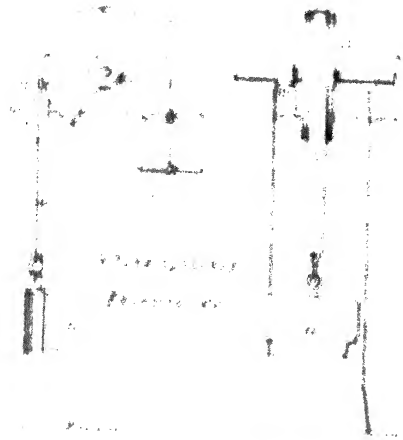


Fig 102

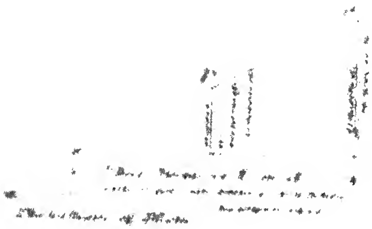


Fig 102a

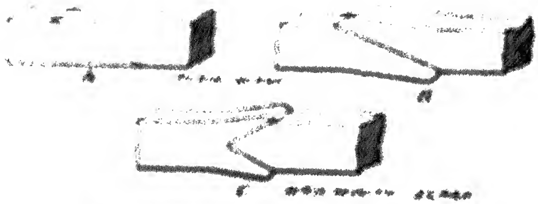


Fig 103 Scarfing

resorted to, the crystallised portion will be left weak and little better than cast iron. This should be carefully noted in making connecting rods of steam engines, or indeed any article the breaking of which might cause danger to life.

Welding.—Wrought iron cannot be cast,* but it can be welded without difficulty; that is, it may be joined piece to piece by heating and hammering, and work of great intricacy may thus be formed. The welding temperature for wrought iron is reached at about 2800° Fahrenheit, and the two pieces to be welded are heated to this temperature, which is detected by the iron beginning to throw out sparks. Two points have to be noticed. The iron should be, if possible, drawn out so that a *scarf* may be made, when welded; this is shewn at A, Fig. 103, and, as will be seen, a greater surface for welding is thereby presented. But, if it be drawn out too fine, it will burn away when put into the fire for the welding heat, and to prevent this it should be left rather thick at the ends, as at B; the lump may be easily levelled afterwards. The two pieces to be welded should both be at their proper heat at the same time, which the smith ensures by changing their positions in the fire, so as to advance the one or retard the other. Withdrawing, he sprinkles them with sand, which forms a siliceous film or flux, and prevents scale by oxidation. Putting them now together, the smith gives one or two blows to fix them, and he and the striker then finish by rapid alternating blows. If the flux be carefully expelled and the joint well hammered while hot, the bar will be nearly as strong there as at any other section. Borax is used as a flux in steel welding. (*See Appendices I. and III., pp.* 748 *and* 917.)

The scarf weld is the one most commonly practised, but the fork weld at C, Fig. 103, is often introduced for large work on account of its greater security.

Having thus briefly mentioned the operations of heating and welding, we shall now proceed to describe the forging of a few objects.

The making of a **Bolt** with hexagonal head is shewn in Fig. 104. A round bar A is taken, of suitable length; it is heated at one end, and *jumped* or *upset*, namely, is lifted by

* See note at end of Chapter III.

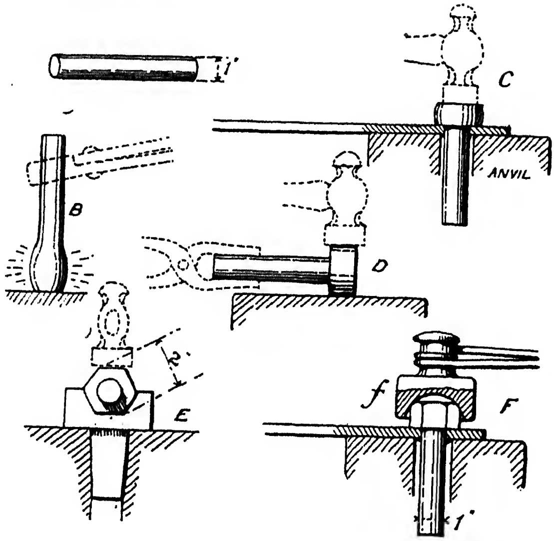


Fig. 104. Forging a Bolt.

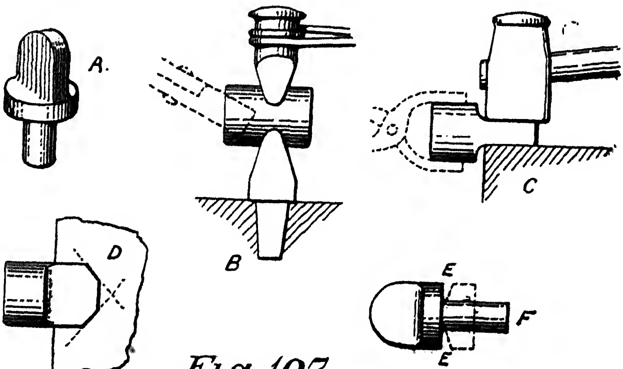


Fig. 107.

Forging a Single-eye.

the tongs and struck on the anvil as at B. A heading tool is next held over a hole in the anvil, and the piece B is reversed and dropped through the tool. Being prevented, however, from passing quite through, on account of the shoulder just formed, it is now beaten by the hammer until the head C is formed. The bolt is then taken out, and the portion C is roughly hammered into the form of a collar at D. It will now have become cold, and must be re-heated to finish the head, which is done in the hexagonal swage E, side after side being presented to the tool by turning the bolt round, and hammering each time. Finally, it is dropped into the heading tool once more, as at F, and, after receiving one or two finishing blows, a cupping tool *f* is applied to give the spherical chamfer.

We may now make a **Nut** for the above bolt. Of course, it is almost unnecessary to state that bolts, nuts, and rivets are now made entirely by automatic machinery, and these examples, therefore, are only intended as an introduction to more difficult forging. A nut can be forged in the form of a ring, and thus dispense with after-drilling. This is the case we shall consider. Fig. 105 illustrates the different operations. Slightly scarf the bar A, which is to be bent round to form the nut, and must, therefore, have the same width as the latter; for example, a three-quarter inch nut would require a bar about three-quarters of an inch by three-eighths of an inch in section. Next heat the end of the bar and bend round the anvil as at B, nicking it through with a blunt chisel (as shewn at *a* in sketch C). Now, put it back in the fire to get a welding heat; take it out; and, breaking off sharply at *a*, lift up the ferrule remaining, on a mandril D, and weld the two scarfings together; then finish the hexagon in the swage E. The nut is not yet complete, however. Re-heating, it is cupped at top and bottom as at F, and the hole is finally made to exact size by the finishing mandril G, which is driven through the nut into the hole *h* in the bottom cupping tool. The nut may now be removed and cooled.

Fig. 106 shews the making of a **Holdfast** for pipes, or pipe hook. Two heats are necessary. In the first a bar is taken, as at A, and is drawn to a 'square' point on the further edge of the anvil as at B, a turn of 90° backward and forward between each

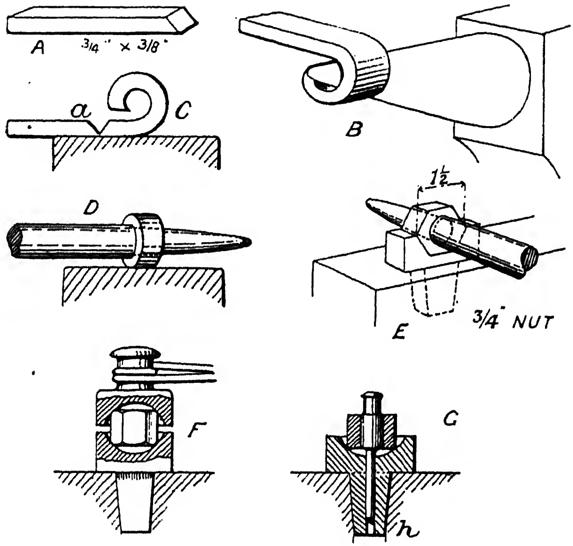


Fig. 105. Forging a Nut.

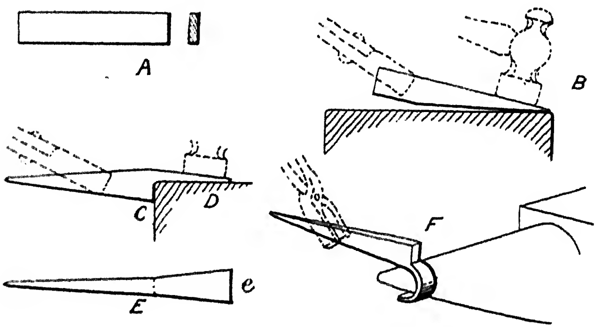


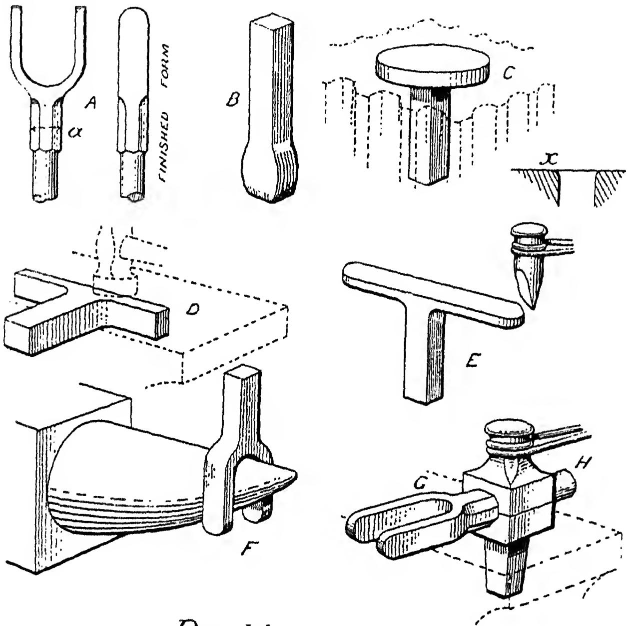
Fig. 106 Forging a Pipe Hook or Holdfast.

blow being given by the hand holding the tongs. A second hea is now taken, and the length of point having been marked off (c the remainder is set down at D, on the edge of the anvil. Here again the bar is turned backward and forward to finish the edges in plan E, and the end is chipped off at *e* to proper length Before the work is too cool the part *e* must be bent round the beak of the anvil, as shewn at F, when the holdfast is complete.

A **Single-Eye** in the form of an eye bolt is shewn finished at A, Fig. 107 (page 103). The hole is to be drilled out afterward. A short piece of round bar is first taken of the same diameter as the collar, and after heating is fullered at B, and set down as at C. On the second heat the edges are hammered, and the corners chipped off with chisel as at D, shewn in plan. One end of the eyebolt is thus finished. Taking a third heat the line E E is marked off, and the tail of the bolt swaged down at F. Finally, the shank is cut off to the required length.

We will now describe the forging of a **Double-Eye**. A in Fig. 108 gives the finished form, serving as the end of a tie or connecting rod, to which it is welded when required. A square bar is taken (exact length of no importance) rather thicker than the part marked *a*, and is first heated, jumped, or upset as at B, and then flattened out in swage block till it assumes the form shewn at C. Being heated a second time it is drawn out as at D, partly on anvil, and partly by returning it to the hole in the swage block, when it is finished off at the ends by chipping off the corners shewn at E. A third heat is required to bend the T thus formed round the anvil beak to the fork shape F, and the fourth and last heat will serve: first, to hammer out the octagonal portion; and, second, to swage out the round part H.

A **Pin** with cotter is our next forging. After heading at the first heat, like the bolt in Fig. 104, it is then of the form B, Fig. 109. On the second heat it is cut to the length required, and the cotter hole marked off. The latter is 'drifted' through by means of the tool C—first, with the work lying in a bottom swage; and, second, to finish—by driving the tool through, over a hole in the anvil, *see* D. In punching and drifting the tool must be kept cool by taking it out of the work, and dipping in the water tank from time to time. A represents the finished pin. The



Double-eye.
Fig. 108.

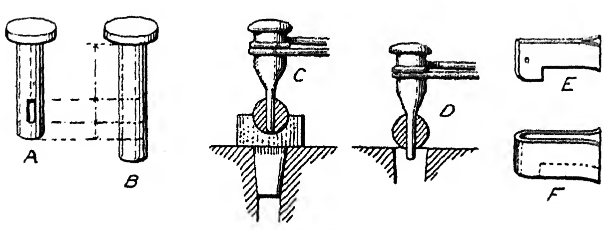


Fig. 109. Pin & Cotter

cotter E needs little description. It may be formed by bending a thin strip of iron as at F, welding the portion near the bend, and chipping out the narrow shank.

The student will have already noticed that a good deal of judgment has to be exercised by the smith in deciding upon the length and breadth of iron necessary to execute a certain piece of work, and, although this can rarely be achieved with very great nicety, yet practice enables him to guess it with sufficient accuracy. As a rule the cubic contents, or the weight of the stuff should be about the same in the rough as in the finished piece, some allowance being made for burning away in the fire, but it is best to err by having rather too much than too little, and in most articles the extra stuff can very easily be cut off. Some, however, require more exact measurement, as from the nature of their construction the after cutting cannot well be resorted to. Wherever parts have to be afterwards machined extra material should be allowed, say from one-sixteenth to one-eighth of an inch, but the careful smith will always leave as little as possible, and if he is directed to finish '*black*' he should make the work as exact to dimension as his tools will allow.

Except in the case of the nut at Fig. 105, none of the work already described has called for the operation of welding. We shall now, however, pass on to some examples requiring the aid of this important process.

A common **S or Double-Ended Spanner** is the article we shall first consider. A, Fig. 111, shews the finished forging. A bar is taken of the same length as the arm, leaving a little extra material for welding. It is heated and first bent to the S form (B) on the anvil beak, straightening by flat hammering on the face of the anvil; and is next drawn out at the ends as at C. Now, two pieces of rather thicker bar being procured to form the jaws, these are heated and bent round the beak, and the corners chipped off and rounded as at D. Heating again, these jaws are finished on the bottom tool and scarfed down as shewn at F. We are now ready to complete the spanner by welding the jaws to the arm, at the scarfings already made (see G), and finish may be given between the flat face H, and the anvil.

In Fig. 112, A represents a **Shackle** for use with chain or

rope. Some little care should be exercised in gauging the length. For an 'inch' shackle, made out of round bar one inch in diameter, a length of fourteen inches would be required. This

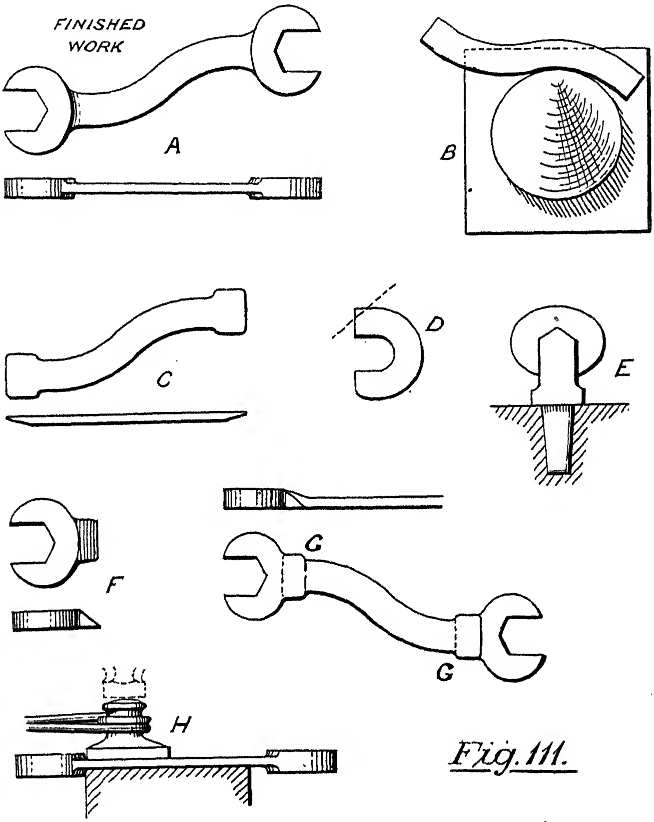


Fig. 111.

S or Double-ended Spanner.

bar is set down to the form at B, by using the set hammer and bottom swage *b*. Two heats, one for each end, are required for this purpose. Another heat for each end enables us to make a

scarf of the form shown at C, by drawing down at the point and sides. The eyes are next formed by taking a welding heat, bending round a mandril rather smaller than the finished size of hole, and welding with hammer as at D. A flat face, E, is used to smoothen, and a finishing mandril is driven through the hole

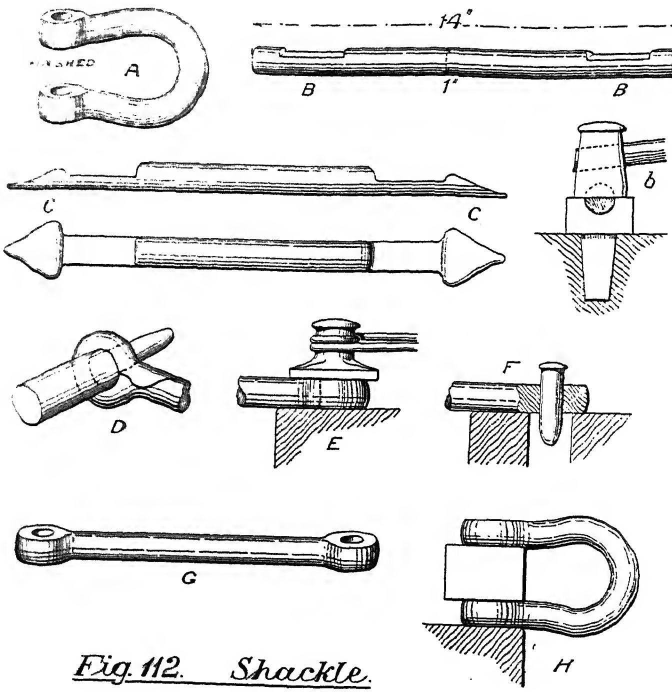


Fig. 112. Shackle.

as at F. These operations being performed on each eye, we have the shackle advanced to the stage G. Only one more heat is now necessary to bend the rod to the proper form round the anvil beak, and the finishing stroke is given on a block (H) which serves as a template to define the distance between the eyes.

An **Eyebolt** of large dimensions is treated in Fig. 113. A is the finished condition. It is such an eye as would be required for the attachment of a rope or chain, being made of round section to prevent cutting or chafing. Here we may begin by taking a round bar of the same section as the part A, and, wrapping it round, scarf and weld it to the form of the eye as at B, at the same time scarfing down the joint again. This done, a second bar C of thicker section is cut to form the shank, and, after scarfing, is welded to B, giving the appearance D. Lastly, the collar is put on by taking a piece of square bar of small section, which may be wrapped round the shank at welding heat and scarfed at E. The bolt is then finished off by fullering the part *b*, and swaging *e*, a rough file being used with advantage afterwards.

Another and probably quicker way of making the eyebolt is to take a bar of the same diameter as the collar, and work out of the solid by swaging down the shank, fullering and flatting out the eye portion, the hole being punched and rounded off.

As an interesting example of punching and swelling out we may take Fig. 113*a*. Here we have a portion of a **Harrow-frame**, and it is desired to form the socket for a common square tyne. The bar at A is first upset, punched, and drifted to the form at B. It should be noticed that at first only a narrow, long section of drift is used, to avoid breaking the bar. The narrow hole is swelled into a round one by a suitable tool on the next heat (shewn at C), and the final step is the further swelling by square drift, as at D, carefully finishing with a flat-face.

Hooks may have the eye formed in the manner described for the shackle of Fig. 112, or the large end may be 'jumped,' and worked from the solid by means of a flat-face tool, either in the case of hook or shackle, and the hole left to be punched or drilled cold. The solid method needs no special description. Assuming a case similar to the one previously described for the shackle, the bar being first round and of the diameter of the thickest part required, the eye end of the bar is drawn to the proper diameter for that place, while the opposite end is drawn down nearly to a point. This is shewn by sketch A, Fig. 114. The eye is next turned and welded, and the hole finished with

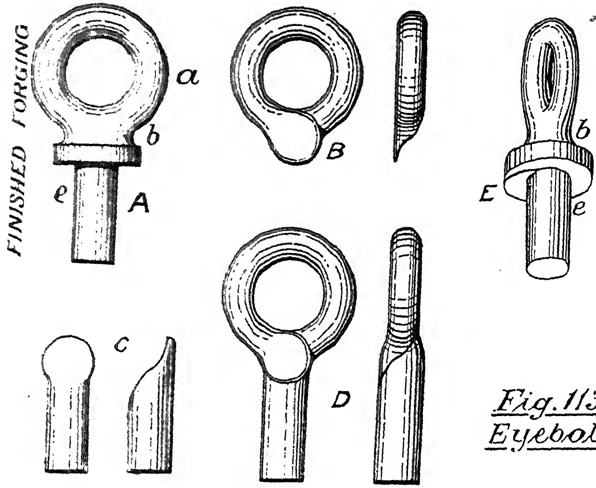


Fig. 113. Eyebolt.

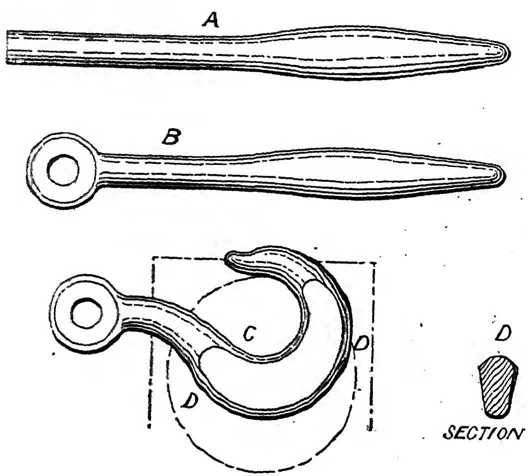


Fig. 114. Hook.

mandril either now or afterwards (B). Heating the rest of the bar the hook is bent to the correct form round the anvil beak C, being constantly checked by rule and sheet iron template; and the proper section given at the same time (shewn at D D) by means of set hammer or flat-face. Both these last-named operations must go together, for the form of the hook will be more or less spoilt by flattening to the section at D D, and this must be again restored by bending.

Bolts in machinery are sometimes placed in very extraordinary positions, so that the spanner in Fig. 111 may have to be discarded, and the **Box-key** (represented in Fig. 115) used in its place. It has a socket at A to fit the nut, and a shank at B, on which a wrench (sketched at C) is placed when required. The key is forged by making the A and B portions separately, and afterwards welding them together. Thus, part A is made by bending a strip of iron, which has been previously scarfed at the ends, into the form of the hollow cylinder D. This is done on the anvil beak, and a second heat is necessary to weld it. The piece B is next formed from a round bar of sufficient section to give the square when flattened. It is shouldered on a swage as at E, sufficiently small to fit into the ring D. And now the small end of E and the cylinder D are both heated to welding temperature; then, being put together as at F, are riveted by striking the mandril G, and by hammering round as at H. The fourth heat is required to work out the square J with flat-face and anvil, and on the fifth and last heat a mandril, which may be hexagonal or square, as desired, is driven into the cylindrical portion K, and the outside hammered until the requisite shape is given to the hole. Removing the mandril the key is considered as finished.

Tongs, having to be used almost continually, are soon burnt away by the fire, and the smith must be able to forge them as needed. We will therefore describe the forging of the round-nosed tongs sketched at B, Fig. 98. The 'bits' that grip the work are made first. For them a piece of square bar is to be set down on the edge of the anvil until it receives the form A, Fig. 116; the successive operations for this are shewn at 1, 2, 3. The two bits should not be made right and left-handed, but exactly alike, for in turning one round axially it will be found to accommodate

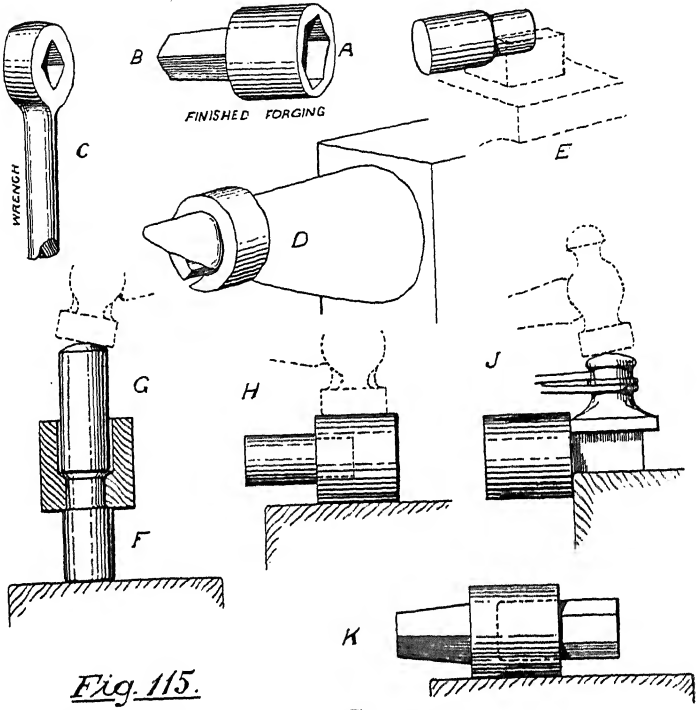


Fig. 115.
Forging a Box Spanner.

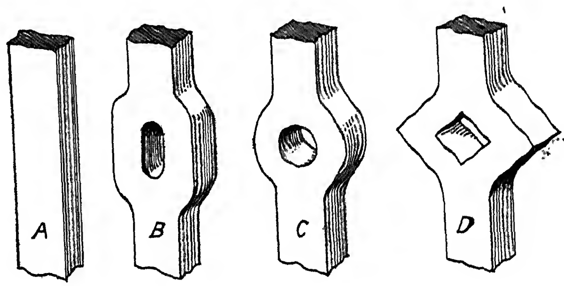


Fig. 113a. *Swelling a Harrow Frame.*

itself quite correctly to the other. One heat should be given for each of these settings down, and during the third the hole (B) is punched. Next, the handles are to be welded to the bits, and for this purpose round rods of sufficient length are scarfed, heated to welding, and united in the usual manner C, being finished carefully in round swages, D. The nose bits are yet flat; they are therefore rounded by means of a top fuller and bottom swage, as at E, and, finally, the two half-tongs are riveted together tightly as at F with a hot rivet, the handles being worked backward and forward while the rivet is cooling, and also during the after quenching in water. This method ensures a well-riveted but workable joint.

The student will notice that in the processes of forging two principal methods are followed, which in many articles merge considerably the one into the other. These are the forging of the object (1) entirely from the solid, by drawing down or cutting out; and (2) the joining of the parts of the forging by welding. The former is a process of cutting out or carving, the latter of building up. Figs. 104, 106, 107, and 108 are examples of the first method, which is the one practised unless the method of welding should be cheaper, and, as we shall see, is always used if possible in large objects that have to sustain important loads. Figs. 111, 113, 115, and 116 are cases where the second method is more useful, for in Fig. 116 a round bar is attached to work that is easiest forged from a square bar, and the end pieces in Fig. 112 are manifestly easier made separately and welded, than they would be by forging completely from the solid.

Further examples of welding are shewn in Fig. 116*a*. In each case A is the work prepared by scarfing or otherwise, and B the built up article. The **Eye** may be said to be merely an example of ornamental welding, for it would be difficult to find a use for it in practice. The **Stud** is more commonly met with; it is prepared as shewn, by scoring the surfaces to be welded with a chisel; less pressure will then be required, the form of the stud will not be so much distorted at the shoulder, and the two pieces are much more likely to enter into each other.

The next three forgings to be described will be worked in the 'solid' manner, and they will conclude our description of those

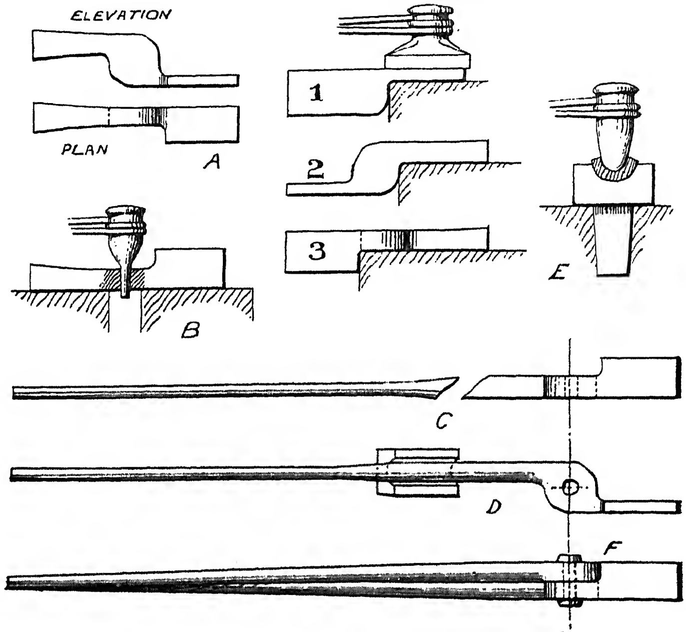


Fig. 116. Forging a pair of Tongs.

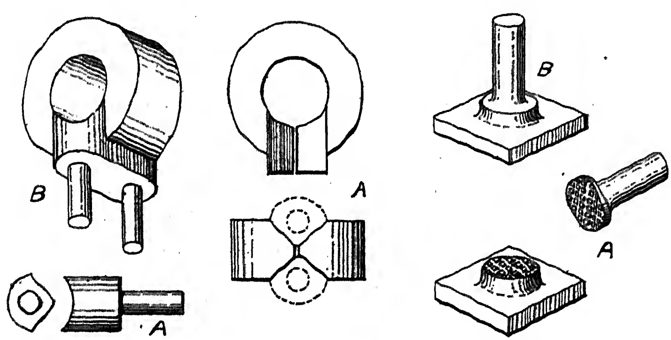
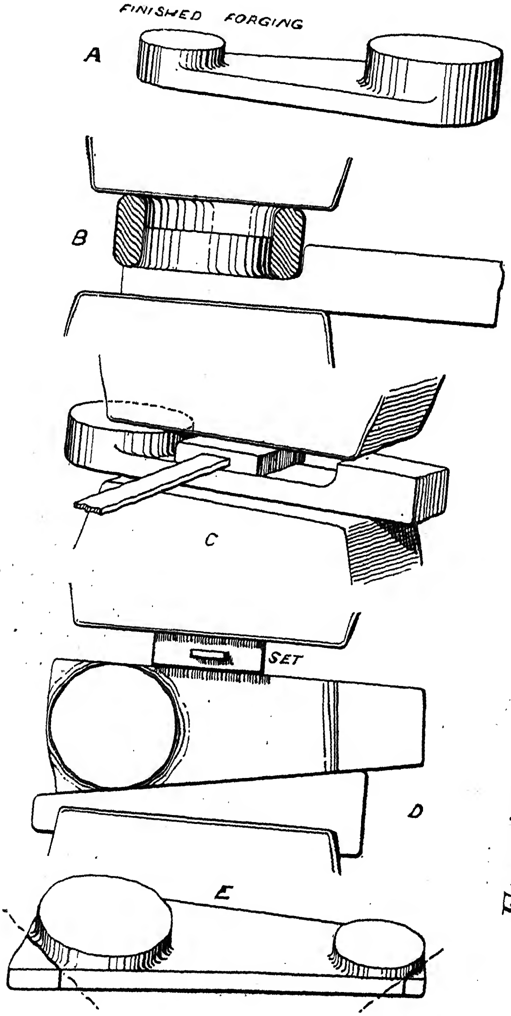


Fig. 116a. Special Welds.

methods used by the smith. They will also introduce the use of the steam hammer, as applied in the smith's shop.



Forging a Single-webbed Crank. Fig. 117.

Fig. 117 is a **Single-webbed Engine Crank**, finished at A. A slab of iron

and width as the largest boss. Heating to a good white heat, it is put under the hammer, where the ferrule B stamps out the shape of the boss. It is next drawn out by suitable tools, called sets, at top and sides (see C and D) until it is of correct length to form the smaller boss, which is first set down to the proper thickness, and then stamped by means of a ferrule, as before. The forging is now of the form E, and all that is necessary is to finish by cutting off the ragged corners round the bosses, which will require another heat—the third; the first having been used for the large boss and the setting down, and the second for the small boss.

A **Bell Crank Lever,** whether large or small, can be made in a similar manner to the foregoing. A, Fig. 118, is the finished lever. A bar is taken, as before, of the thickness and width of the boss. It is first bent to a right angle—if a small lever this may be done on the anvil beak, but, if large, blocks would be put under the steam hammer, with the hot bar between, as at B. That done, the boss is next formed by ferrule, as at C. Another heat will now be found necessary for each arm, in order to set down, as at D D, to proper section, and the ends are finally cut to curve by means of a chisel or cutter (see Fig. 119*a*).

Figs. 119 and 119*a* represent the forging of a **Small Crank Shaft,** say two inches in diameter, such as can be worked by the smith with the aid of a small steam hammer. A is the finished shaft, and has two crank arms forged upon it at right angles to each other, in the manner of locomotive axles for 'inside' cylinders. We must, to begin with, have a slab of iron of square section, sufficiently large to form the crank web when drawn down. This is seen at B. It should also be long enough to complete the whole shaft when drawn down and swaged in the manner to be described. The bar B is first to be formed into the shape shown at C, by heating to a good white heat and setting down under the hammer, as at D. This will leave the slab of the same section as the crank web, and, if carefully set down to the form indicated, the webs will now be in correct position, namely, at right angles to each other. Of course some care must be taken, the right angle being tested with a square, and the part *a b* in particular should be made of such a length that when swaged to the round section it will measure the correct distance

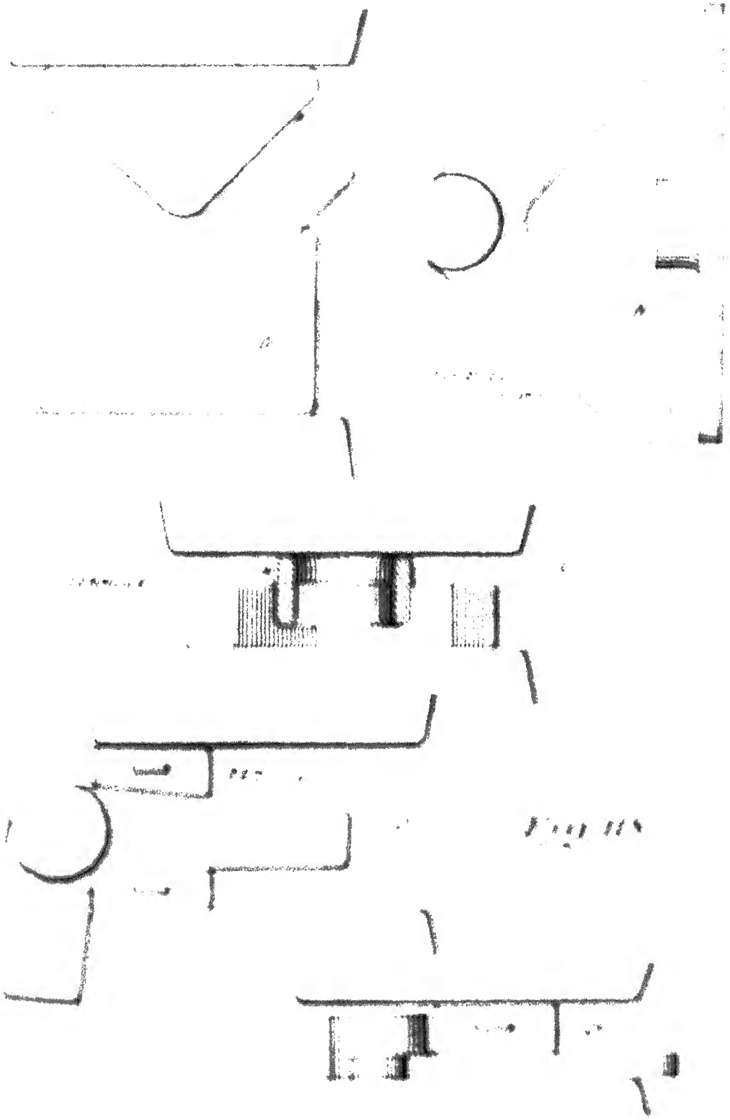


Fig. 118

Bell crank Lever

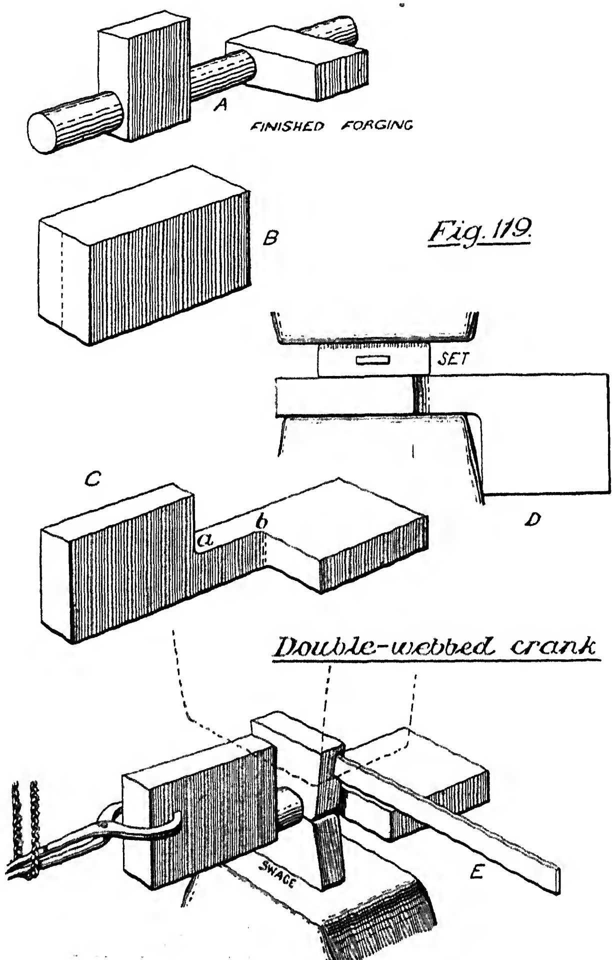


Fig. 119.

Double-webbed crank

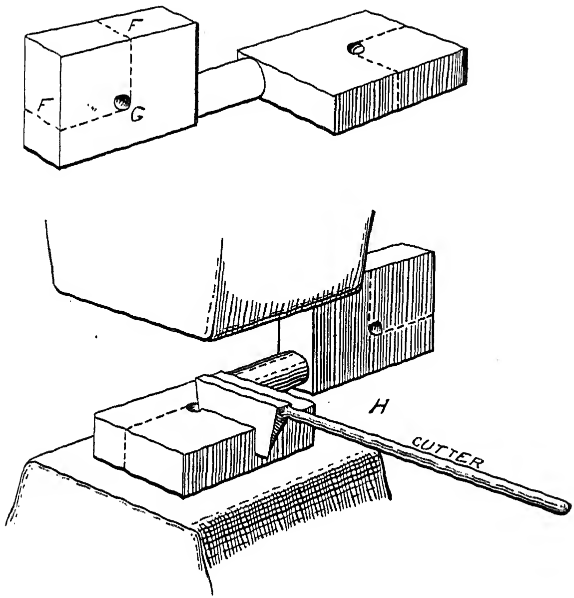
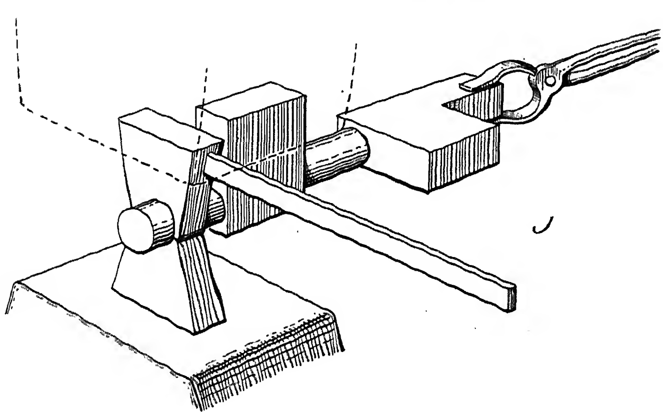


Fig. 119a.



between the crank arms. Probably this piece had better be swaged next (it may require another heat), the forging being turned round, backward and forward, to produce a good result (see E). The distance between the cranks should be now finished very exactly, by knifing or other means. The ends remain. Here it is necessary to first cut out the superfluous material by marking off at F, punching the hole G, and, while the crank is still hot, cutting out the rectangle with a knife or cutter (see H). Afterwards the shaft is rounded by swaging (J). When this has been done for both ends, and the shaft carefully measured, as well as tested for axial straightness, straightening if necessary, the work may be considered complete. In this form of crank (double-webbed) the pair of webs are always forged solid in the manner described, and the piece between taken out either by slotting or turning in the lathe.

At this point we may as well consider one other form of crank, which has many advantages. In Fig. 120, A is the shaft alluded to, and is there shewn finished by turning in the lathe. It is considerably stronger than the one previously described, on account of the fact that the fibres follow the bend of the crank webs (represented in dotted lines), while in the shaft of Fig. 119 these fibres are cut through when the mid pieces are slotted out, which must of course weaken the webs considerably. The only objection to the form here shewn is that a great width is required for the crank itself, and, as this cannot always be spared, the crank has only been applied on portable or traction engines up to the present. Properly we might have described this in the space devoted to the forge, for a larger hammer is required than commonly occurs in the smithy. A bar of the best Yorkshire iron, of sufficient diameter to turn down to finished size, is heated and placed between the blocks B B, and these are made to approach each other by blows from the hammer, at first gently, and afterwards more strongly. Lastly, the shaft must be tested for straightness.

Stamping.—Where several articles are required exactly alike in form and dimension they can often be forged more cheaply by the use of stamping tools. The crank last described might almost be termed an example of this kind of work, and the lever

Use of Bending Blocks. Crank Shaft.

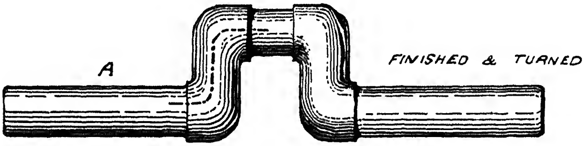


Fig. 120.

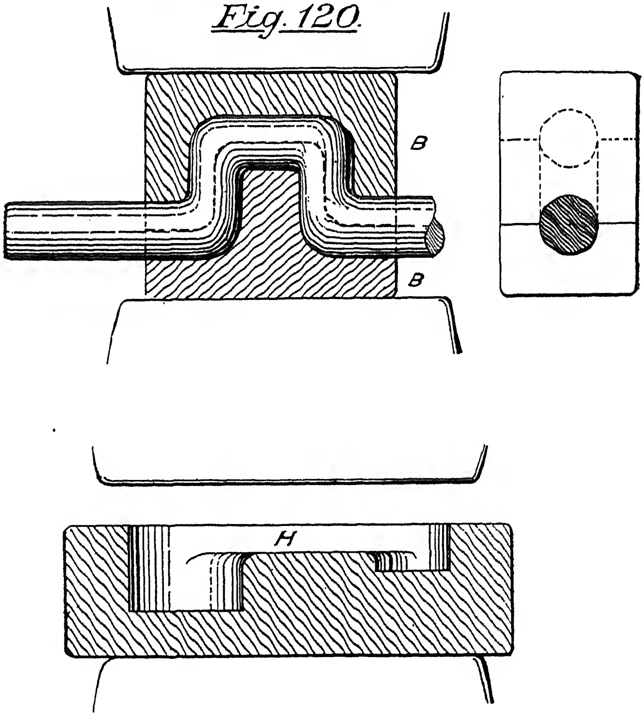


Fig. 121. Stamping Die

in Fig. 117 could be stamped by means of the tool shewn in Fig. 121, the hot iron being placed in the hollow H, and the hammer brought down upon it. The ragged portions are afterwards chipped off the forging. Usually these stamping tools are made of massive cast iron, but if they are to be used extensively cast steel will be found necessary. Other examples of work suitable for stamping are shewn in Fig. 121*a*, where A is a spanner, B a double eye, C the centre portion of a screwing stock, D the handle portion of a lever, and E the boss part of the same lever. (*See Appendix II., p.* 802.)

Before leaving the smithy two processes should be explained, because they are as a rule performed by the smith. These are the methods of hardening wrought iron and steel. Cast iron, as we have seen in Chapter I., can be easily hardened at the surface by chilling, this taking place while the casting is in course of formation. Wrought iron and steel are hardened after the article is completed.

Case-hardening.—This is the name given to the process by which wrought iron objects are hardened to a depth of from one-eighth to three-sixteenths of an inch below the surface. After forging the work is machined and polished, and is then made to absorb carbon by being placed in air-tight boxes or *cases* in contact with some substance rich in carbon, being strongly heated while in that condition. The method is much the same as that pursued in the cementation process (Fig. 88), and it will therefore be seen that the iron at the surface is converted into a film or case of steel, the only difference from the cementation process being that the heat is merely kept on long enough to case the iron with steel and not to steel it quite through. While the iron is left, then, hard at the surface the inside remains tough, and is as capable as ever of enduring vibration. The boxes may be either made of sheet iron, or may be fireclay retorts similar to those in use at gas works, and provided with a lid to keep them air-tight. They may be heated as in Fig. 88, and the substance put in contact with the iron is not wood charcoal, as in cementation, but animal charcoal in the form of bones; for it is found, why it is not quite clear, that if nitrogen be present the carbon will unite more rapidly with the iron. Other substances may be used, such

as prussiate of potash, leather or hoof scraps, but the process is chemically the same. After packing, which must be carefully done, to prevent the articles bending while hot, the heat is raised during two hours, the whole kept at a regular temperature for

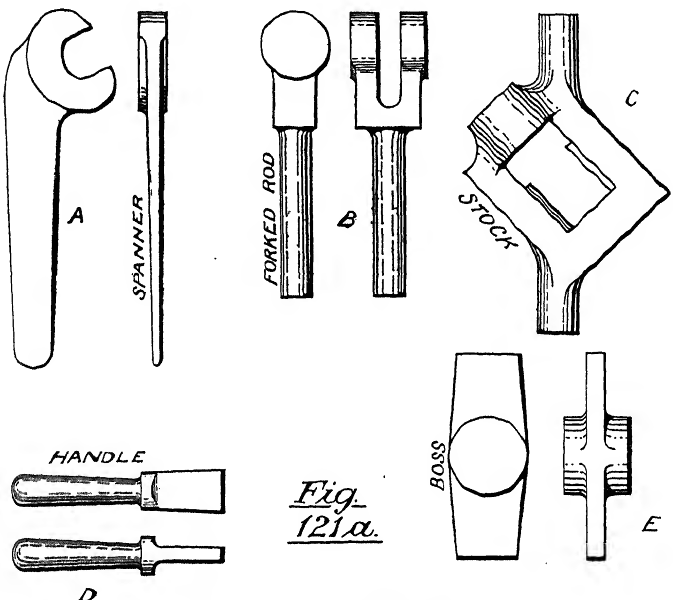


Fig. 121a.

Stamped Work.

about nineteen hours, and then allowed another two hours to cool. Removing the articles they are quenched in water, straightened, and re-polished. *(See Appendices I., II., and V., pp.* 748, 803, *and* 973.)

Steel of a mild quality may be hardened at the surface by the absorption of more carbon.

Such small articles as have to withstand considerable wear are case-hardened, *e.g.*, radius links for reversing gear.

Tempering is a method of giving to a piece of steel any required degree of hardness. Properly there are two distinct processes meant when we speak of 'tempering' a steel tool. The

first of these is that of *hardening*. Here the steel is heated as equally as possible to a 'cherry red,' and not more; and on withdrawing from the fire it is plunged vertically into a vessel of cold water. The quickness of cooling has a great effect on the hardness, and this may be accelerated by moving the article about in the water. Cracking or warping will also be prevented by judicious motion.

The steel is now so hard that it will scratch glass. It must next be *tempered* or let down to the required degree of hardness. If the tool be again heated to cherry red, and allowed to cool slowly it will by that means have become annealed, and will be at its softest; but if it only be heated to one of the temperatures in the following table (Fig. 117*b*, Plate III.), and then cooled rapidly, it will take a particular degree of hardness corresponding to that temperature, and to be obtained at no other. When letting-down, the *softest* tool will be that which is cooled at the *highest* temperature, and the *hardest* that cooled at the *lowest* temperature.

The exact temperature which the tool has assumed is ascertained by the colour which appears on the brightened surface, due to a film of oxide of iron formed by contact with the air. There is some difference of opinion as to the requisite hardness for certain purposes, and slightly different colours are required for different steels, but Plate III. is suitable for average tool steel.

Tempering a Chisel or Drill.—To make the matter clearer we will take the case of a chisel for chipping metal. It is forged out of a steel bar of the section shewn at A, Fig. 122, and is drawn out (at as low a heat as possible, to prevent burning) to a flat point as at B. This point is now to be hardened and tempered, while the rest of the chisel is to remain in its natural condition. Whenever the tempering is accomplished by quenching in *water*, the preliminary process of hardening must always be performed, otherwise the tempering would have no effect. In the case of the chisel, or any tool having a point requiring a particular temper, the two processes are performed at one heat, but it must be quite clear that hardening is not therefore dispensed with. Heating the *whole* chisel to a *cherry red*, the part *a b* only is quenched in water, and so becomes very hard. Now rub the point of the chisel with a stone to brighten it a little, and, as the

Fig. [illegible] Tempering Table

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	COLOUR	TEMP. F.	USE.
VERY HARD	CHERRY RED	1650°	For Preliminary hardening only
SOFT	PALE BLUE	600°	Too soft for anything
ELASTIC	DARK BLUE	570°	Springs Screw drivers Circular saws for Metal Cold chisels for W.I. Firmer chisels
	DARK PURPLE	550°	Cold chisels for C.I. Axes and adzes Cold chisels for Steel
	LIGHT PURPLE	530°	Augers
	YELLOW PURPLE	520°	Flat Drills for Brass Twist Drills Plane Irons
	BROWN YELLOW	500°	Gouges Reamers Punches and Dies
HARD	VERY DARK STRAW	490°	Chasers Taps Screw-cutting Dies
	MEDIUM STRAW	470°	Boring cutters
	LIGHT STRAW	450°	Milling cutters Drills Iron-planing tools
	VERY PALE STRAW	430°	Steel-planing tools Hammer faces Light turning tools Scrapers for Brass

Fig. 117 b.—Tempering Table. ✓

Face p. 126.

heat from the body $b\,c$ travels down towards a, the colours will appear, the point becoming gradually hotter, yellow first, then through brown to blue. But we require for our chisel the temperature of 550°, which is indicated by a dark purple; as soon, then, as this tint is seen, the chisel is entirely plunged into water, and the point is thus made of the correct degree of hardness. A drill point may be tempered in a similar manner, using, however, the darker straw yellow for colour.

Tempering a Screw-tap.—Sometimes, when an even temper is required over a considerable surface, the result may be better obtained by putting the article in contact with a body of hot metal. Such a case is that of a screw-tap. The tool, being finished and polished, is next to be so tempered as to make the

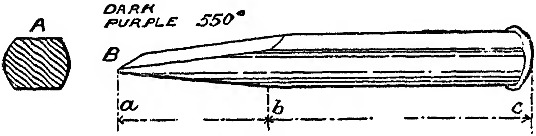


Fig. 122. Tempering a Chisel.

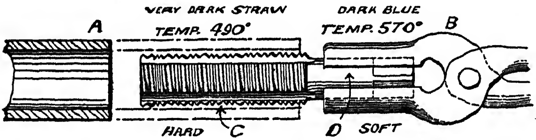


Fig. 123. Tempering a Screw-tap.

screw threads hard, while the square shank remains soft. An iron tube being procured, A, Fig. 123, of such a diameter as to just fit freely over the tap, the latter is first lifted by the shank, within red-hot tongs, B. In the meantime the tube has been heated to a dull red, and as soon as the first signs of straw colour are seen on the tap shank, due to the heat from the tongs, the tap is placed within the tube as shown. The shank will have had, so to speak, a start of the rest of the tap, and by the time C has assumed a dark straw colour, D will have arrived at dark blue, a

higher position in the colour scale. At this point the tool is quenched in water.

Two other methods of ascertaining the desired temperature are in use besides the colour test. These are the flashing temperatures of certain oils, and the fusing points of certain alloys. The first is practised by coating the part of the tool with oil, and holding it over the fire until it blazes off, then quenching in water. In the second, the alloys are usually of lead and tin, and vary from equal parts of each metal to complete disappearance of tin and consequently total lead. A bead of the alloy placed on the tool, may be watched until it melts, and the part then quenched. Of course, as before, the two operations of hardening and tempering are required. Watch springs are tempered by the blazing-off of oil at a temperature of 570°, producing a dark blue.

Hardening in Oil.—This may be looked on as a species of tempering without preliminary hardening. It is of great value when dealing with articles having very large surface, and which could not be heated to an even colour by the methods previously mentioned. Only one degree of hardness can, however, be obtained, that corresponding to a dark straw colour in the table. A pan of oil being provided of sufficient capacity, the article, heated to a dull red, is plunged into the oil; and the softer result when compared with water hardening is no doubt due to slower cooling. (*See Appendices I. and II., pp.* 749 *and* 803.)

Gun cores are cooled in oil, to enable them to withstand the wear due to the shell, and also to increase the strength of the steel. Thus, in some experiments at the Terre Noire works, four specimens of steel were heated and cooled in oil, and it was found that whereas the average breaking stress per square inch was 35·29 tons before the operation, it had afterwards increased to 51·23 tons.

It should finally be noticed that much care is required in tempering—care not to overheat in the first operation; care not to warp the tool in cooling; care not to crack the tool at the water level. Some tools will harden best in a saturated solution of salt, others in a stream of running water. Generally it is wise to move the tool well up and down during cooling. Hardened steel may be compared to *glass*, annealed steel to lead, and

tempered steel to *whalebone.* Our process then when tempering by the aid of water is to raise the steel to 'glass,' and then lower it gently to 'whalebone.' Hardening in oil gives the 'whalebone' without passing through the 'glass' stage.

THE FORGE.

We shall now pass on to describe the turning out of very heavy forgings, which include all articles too ponderous for smith's work, and which are consequently made in the forge under a very heavy **Steam-hammer.** Fig. 124 is a drawing of a hammer suitable for general forge work, such as we are about to consider, but, of course, extra large forgings would require special-sized hammers.

The hammer in Fig. 124, Plate IV., has a falling weight of five tons. After the careful account of the smith's hammer there will be very little to say here by way of description. As before, the outer valve is for the purpose of *admitting* steam (being opened by a screw acting at the end of a lever), while the inner valve controls the *direction* of flow, the exhaust passing upward. The long hand-lever serves to move the distribution valve, and the self-acting arm between it and the valve reverses the latter as soon as the arm is moved by the tup on its upward travel.

The **Furnace** used by the forgeman is very similar to that shewn in Fig. 85. It is there called a Puddling Furnace, and indeed 'blooms' are to be made for heavy forgings just as in the case of puddling, the only difference being that they are built from scrap iron instead of white pig. A pile of scrap iron is heaped on a rough wooden tray, and is then put into the furnace. Several of these piles being so placed and heated sufficiently, they are then found stuck together. Withdrawing them, thus adhering, by means of very large tongs having a balance-weight on the handle end, and supported at the middle by a crane, the blooms are put under the hammer and well beaten together to form slabs. It will be these slabs that we shall use to build up our forgings. Fig. 125 shews an arrangement of furnace and cranes for heavy forgings.

First we shall consider, in detail, the forging of a **Double-throw crank shaft** of large size, the finished form of which is

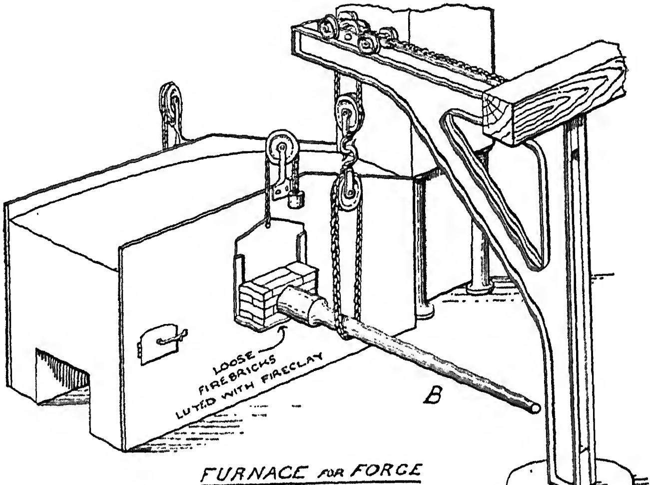


Fig. 125.

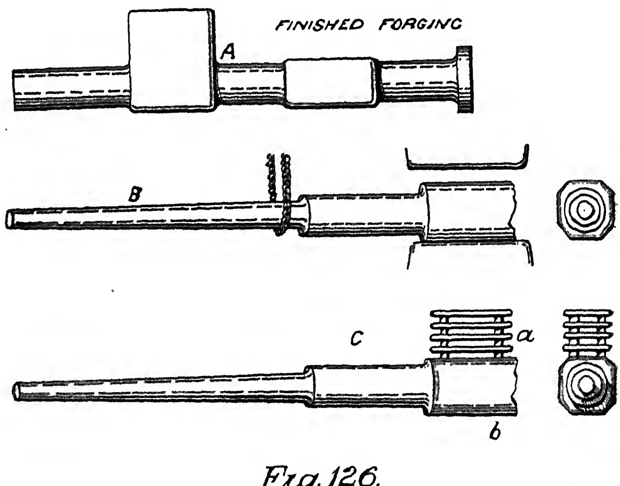
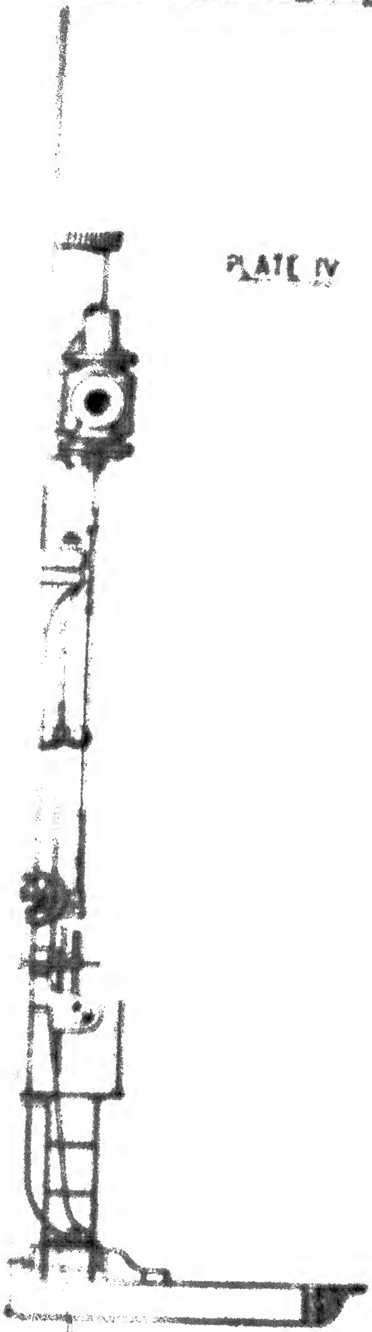


Fig. 126.
Large Crank Shaft.

PLATE IV



seen at A, Fig. 126. The forgeman always requires a staff or '*porter*' to carry his forging, to which, for the time at least, the latter is welded. It is simply a long tapering bar B (Figs. 125 and 126), supported by a crane chain, and carried to and from the furnace by the undermen, while the head forgeman directs the hammerman, and applies the different tools to the work under the hammer. The end of the porter is put in the furnace and made to pick up, at a white heat, a few slabs which have been previously placed there; putting them under the hammer they are all thoroughly welded, and the round form of the first part of the shaft obtained by swages similar to those of the smith, but of suitable size. More slabs are added, and welded, until the shaft is sufficiently long to take the first crank web. The web is now built up by laying slabs upon it as at C (Fig. 126), the end being put back in the furnace. Care must be taken in piling these slabs, both now and always, that space be left between them by the placing of pieces of scrap, so as to enable them to take a welding heat right through. Bringing the hot slabs back to the hammer, they are welded by striking both at top and sides: and so the process is repeated on both sides, *a* and *b* (Fig. 126), until the shaft has the form D (Fig. 126*a*). It is then set down as at E. But the web is not yet finished. Heating again, it is flattened out to the shape F, and slabs are again piled on and welded to the body of the material, the process being repeated as before for both sides of the web. The object of laying the slabs on both sides of the web is to keep the direction of the fibre such that the crank may be best suited to meet the stress put upon it. By this time the forging, being unbalanced, will be difficult to turn round; but this is overcome by clamping four arms *d d* on to the porter, these being turned by the strength of two or four men as required. The web is now hammered at top, bottom, and sides, to correct dimensions, the ragged end *e* chipped off by means of a cutter, and the other end *f* cut down with the same tool, the extra piece G (Fig 126*a*) being worked by sets until drawn out to receive more slabs. The shoulder *g*, and the piece G, are next finished to the round by means of swages, and the building of the second web commences. This is carried out in exactly the same manner as the first one, except that it must

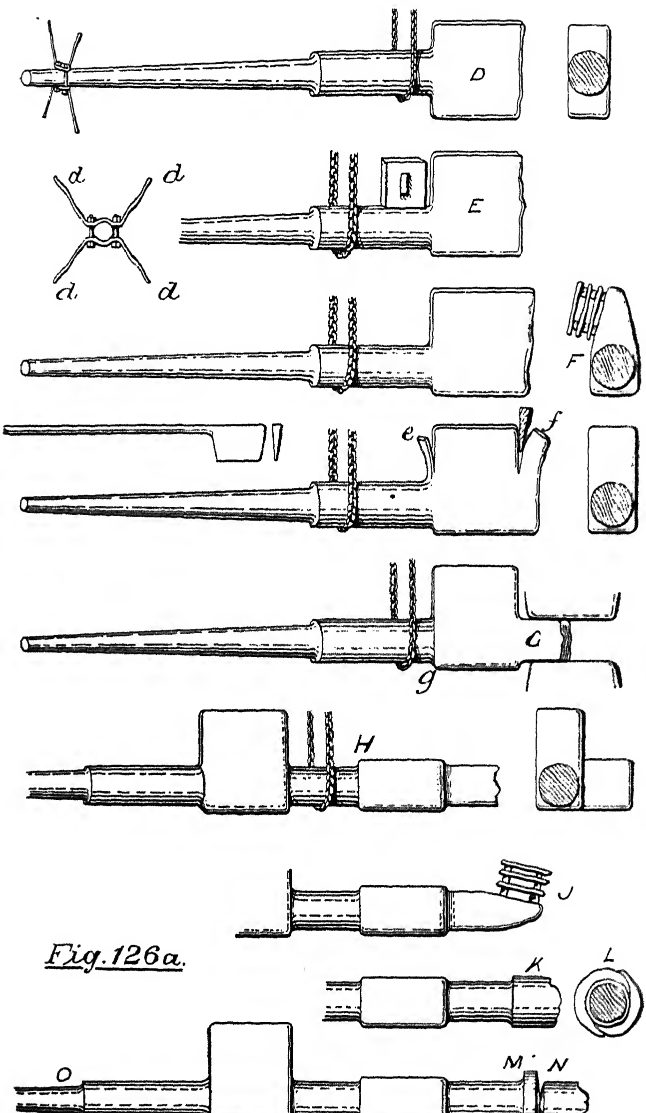


Fig. 126a.

Large Crank Shaft (continued)

be carefully built at right angles; this point, as well as that of the general straightness of the shaft must be gauged with square and straight-edge by the head forgeman, as the work progresses.

By this time then our forging has reached the condition H, and as the sketch A, Fig. 126, shews us a solid collar, for the purpose of coupling to another shaft, we must add this portion. Slabs are again piled up as at J, Fig. 126*a*, heated and welded, until sufficient stuff has been worked together to form a small collar K, and then the whole collar can be finished either by the slab method, or scarfed bars (L) can be wrapped round the shaft and thoroughly welded. Finally the collar can be chipped down at M to the correct length, and cut off entirely at N. There only remains the porter end O, which may be finished by taking off the handles, and clamping them at the collar end, then putting the porter through the furnace till it protrudes at the further door, and after heating cutting it off to the length shewn on the drawing. The shaft is then set aside to cool.

Steel Shafts are forged from ingots (obtained by any of the processes mentioned in Chap. III.), and being thus treated from a solid block, differ in no sense, except size, from the example shewn in Fig. 119. Some makers prefer, after flattening the ingot to the thickness and height of the crank webs, to set down the central portion of the shaft, forging each web in the same plane; and afterwards, to turn one web at right angles to the other by twisting the shaft; but there can be little doubt that this is an objectionable method, and should never be resorted to. A good deal of care, in the case of steel, should be taken to get rid of the blow-holes previously mentioned as existing in the ingots, and as simple hammering is usually insufficient, *cogging* is the operation performed, which consists in partly punching the steel while hot immediately over any portion where honeycombing is suspected—a sort of kneading, in fact.

After the careful description of the crank-shaft forging, a short explanation will suffice for the following articles—*Piston-rod* with *Cross-head*, and a *Connecting-rod.* Whenever such forgings are made of wrought iron they are built up from scrap as in the case of the shaft, such scrap consisting of all kinds of wrought iron,

especially the shearings of plates from the Boiler Yard, and this being worked over and over again in the manner previously described we naturally obtain a better quality of iron than that which has been but once puddled. Another point to notice is that the slabs should all be perfectly welded by good hammering *before* the forging is actually formed to the required shape, for much working after cutting to proper dimension will cause distortion; while if, on the other hand, sufficient hammering is not given to the slabs, cracks are sure to show after machining, and the piece will be dangerously weak.

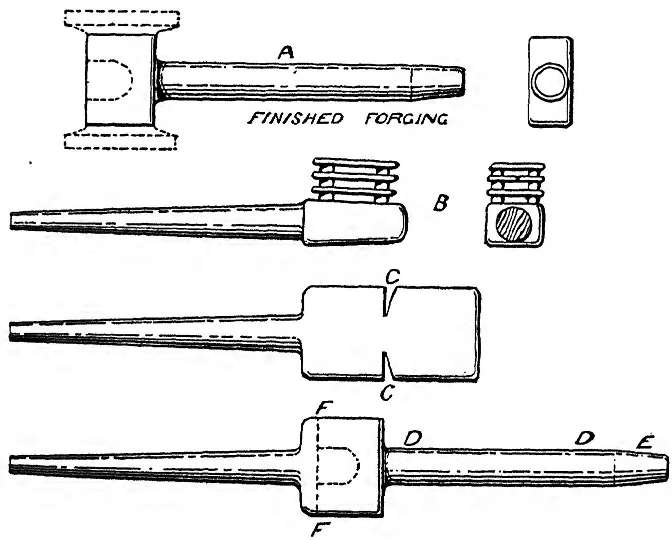


Fig. 127.

Piston Rod.

Fig. 127 will serve to shew the forging of the **Piston-rod**. Its finished form is given at A, the cross-head being solid with the rod, and having renewable 'slippers' of cast iron. Slabs are piled on the porter to form the cross-head, as at B, first on one side and then on the other; sufficient, if possible, to complete both

cross-head and rod. The shoulder of the butt is next knifed out at C, and the rod drawn down and swaged as shewn at D, the taper given at E, and the whole cut off to proper length. Finally, the porter is put through the furnace, as in the case of the crank shaft, the clamps being transferred to E, and the butt end finished by cutting off at F to the correct length.

The **Connecting-rod** in Fig. 128 is a little more difficult,

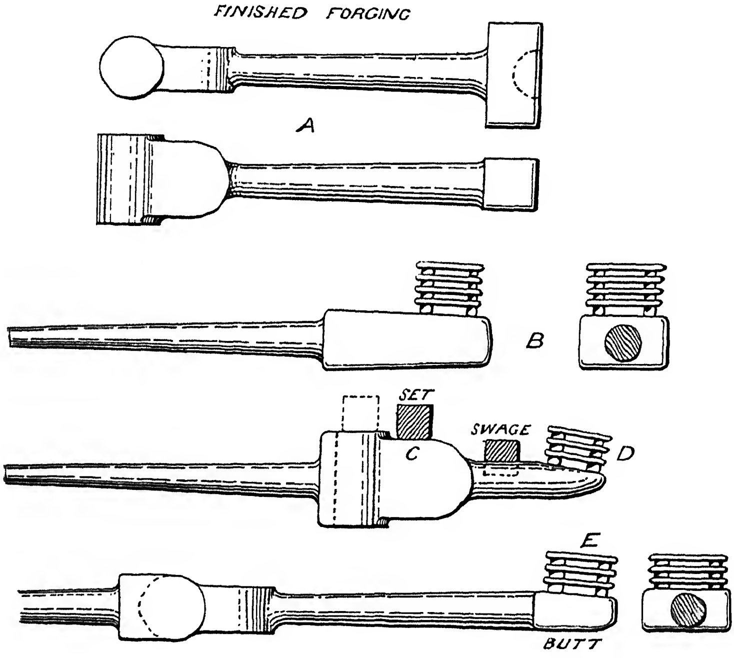


Fig. 128.

Connecting Rod.

but no new principle is involved. A is the finished rod. Sufficient material is first attached to the porter to make the forked end and about half or more of the rod. This is shewn in progress at B. It is next drawn down by sets and swages to the form C, and

more slabs are piled on to complete the rod and butt (*see* D and E). No further description will be needed to finish the forging, as there only remains the cutting off of the butt end to correct dimension, and the severing of porter from forging as in the previous examples.

Want of further space compels us to close our chapter on Forging, but sufficient examples have no doubt been given to stimulate the student, who will now without difficulty be able to construct other forgings for himself, albeit more complicated than those already given. Of course different workmen have slightly different ways of arranging their material, and no two will exactly agree, but that forging will be the best one where the fibre is disposed so as to meet in the best way the stress coming upon it.

Hydraulic Forging.—*See Appendix II., p.* 803.

Cold Drawing.—*See Appendix II., p.* 805.

CHAPTER V.

MACHINE TOOLS.

THE pattern maker, moulder, and smith having supplied us with rough castings and forgings, it is now necessary to finish these articles truly before passing them on to the erector. After marking or measuring-off, certain portions of metal have to be removed by hand or machine tools. The remainder of our work will then consist of—*Marking-off*, or indicating the finished outline by a boundary mark; *Machining*, or removing superfluous material by automatic or semi-automatic machine power; and *Fitting*, which is the finishing of certain parts by hand power, usually the chisel and file.

Machining has always tended to gradually usurp fitting by hand, and its advance is so rapid at present as entirely to take the place of handwork for such articles as are to be repeated; in such instances manufacturers have special machines designed. Even in unrepeated work a much larger quantity is done by machine than hitherto, perhaps most of all by the extended use of such tools as milling machines.

As so much depends on the perfection of a machine tool itself (the workmen merely 'setting' the work and arranging speeds), a thorough knowledge of these machines is necessary, so as to appreciate their capabilities and enable us to design work to suit them.

The next chapter has been reserved for the *operations* of marking-off, machining, and erecting, the present being devoted to the **Machines** themselves, which may be classified as *Lathes; Planing, Shaping and Slotting Machines; Boring and Drilling Machines;* and *Milling Machines.*

Of course there are many *general* varieties of each class, and each variety is again varied to suit special needs. Thus, as regards drilling machines, most inland workshops are supplied

with *conical* [illegible] for drilling [illegible] stand under a [illegible] usual [illegible] has [illegible] largest [illegible] than [illegible] generally have [illegible] a hand [illegible] moving [illegible] shaping machines.

It will be seen, therefore, that [illegible] adequately describe [illegible] *imperative* [illegible] only [illegible] of the four before-named [illegible] and [illegible]

General Principles. [illegible] lathe was the earliest invented [illegible] principles of [illegible] as a reciprocating machine [illegible] revolved in either direction alternately, [illegible] and tool to give it continuous rotation, and the *forward* [illegible] only was effective, the *backward* revolution being lost. Planing, slotting, and shaping machines are present examples of this early stage, as far as reciprocation is concerned. [illegible] give these tools a continuous action [illegible] tools, have lately been [illegible] doubt the difficulty is to be surmounted [illegible] the milling machine, corresponding [illegible] evolution to that of the circular saw, band saw, and wood planing machine, which all began as reciprocating [illegible] (See *Appendix III.*, p. 418.)

The milling machine has [illegible] the cutter, which is also troublesome to sharpen and difficult to keep to proper profile, and the [illegible] The latter objection has been met by [illegible] easily re-ground, the stock being retained.

In the lathe, then, we have a stationary tool and revolving work generating a *cylindrical surface*, while the milling machine has a revolving tool and stationary work moving [illegible] the end of its

feed) a *plane surface;* and nothing could be more satisfactory. If this be the completion of the cycle, as we suppose, then the reciprocating tools, with lost back-stroke, must ultimately give way.

The Copying Principle is another great principle involved in both hand and machine tools. All depend for their accuracy on one or more carefully-prepared *copies* contained within the tool. Thus in the carpenter's chisel the flat back is held against the wood when paring, and constitutes the copy. The sole of a hand plane serves the same purpose, its truth or otherwise being copied on the work, which may be proved by curving the sole, and thus obtaining curved surfaces.

The copying principle is universal. Take the lathe : the bed has a plane surface truly parallel to the line of centres, thus enabling us to produce a true cylinder as our solid of revolution. A second slide at right angles to the former gives us a copy for use in 'surfacing,' producing plane ends or *right* cylinders.

The V grooves of the planing machine give accuracy along the table, while the cross beam or slide ensures truth across it, and so we obtain a true plane. The vertical slide and the two horizontal cross slides are the copies in the slotting machine, while the shaping machine has two copies supplied by the horizontal slides, at right angles. Lastly, the milling machine has two slides, at right angles and also horizontal.

As the truth or otherwise of these copies is transferred to the work, it is of the utmost importance that they should be made perfectly correct in the first instance.

The copying lathe and other duplex wood-working machines are further examples of the principle. (*See Appendix II., p.* 812.)

✓ **Cutting Tools.**—We will now consider the shapes and angles required for the tool itself. As a rule wood-working tools act by wedging, or splitting-off the shaving ; and the resistance is tensile, with some bending. Our interest is with cutting tools for metal, and Prof. R. H. Smith has shewn their action to be totally different.

The diagram Fig. 129 represents the tool in action. B is the

angle of relief or clearance angle, to keep the tool clear of the work; A the *cutting angle*, and C the *tool angle*.

The point C requires great strength for metal tooling, and as this makes A very large, 'paring' cannot occur, but the material will be 'crippled,' either by compression, shear, or a combination of both. Sections parallel to F G will be in compression, and those parallel to E G in shear, and it will be evident that along

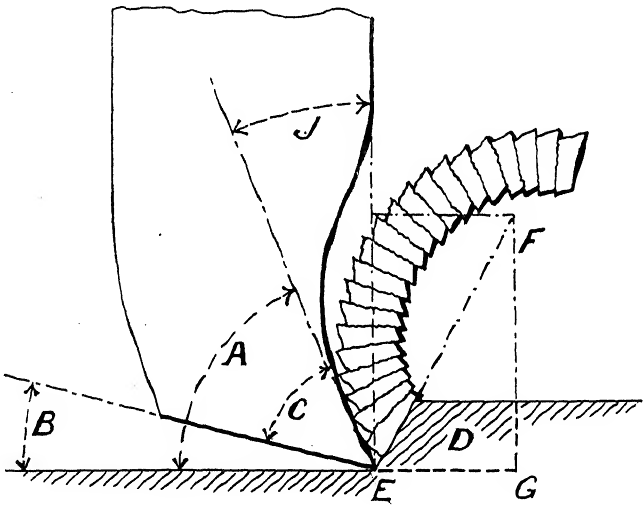


Fig. 129.
Action of tool in cutting metal.

some section E F, the material will be weakened to the greatest extent; here then the shaving breaks so much as to curve up the face of the tool. The direction of E F will depend on the relative values of the compression and shear strengths of the material.

Great heat is generated, due to molecular resistance and friction. A lubricant of soap and water* is used for ductile materials like wrought iron, contained in a can placed above the tool-box, and led to the tool point by a wire, down which it

* Or soda, if rusting is to be avoided.

trickles. This cools the tool, and lessens the friction between tool and shaving. For cast iron and brass these precautions are not needed.

There has been, up to the present, some diversity of language regarding the angles A, B, and C (Fig. 129). Thus, in the planing tool, A has been termed the cutting angle, while in the lathe tool C has been so called. Manifestly the first is the more reliable nomenclature; then C may be called the angle of the tool.

Their values were determined by Hart thus:—

	For cast iron.	For wrought iron.	For brass.
A—Cutting angle	54°	55°	66°
B—Relief angle	3°	4°	3°
C—Tool angle	51°	51°	63°

This supposed the least force of propulsion was required. But if endurance of point be considered, a larger angle is usually given, as follows:—

	For cast iron.	For wrought iron.	For brass.
A—Cutting angle	70°	65°	80°
B—Relief angle	3°	4°	3°
C—Tool angle	67°	61°	77°

In a lathe tool B is termed the *bottom rake*, and J the *top rake*, while a third angle with top of tool, but on right *or* left side, is called *side rake*.

These angles will serve for any machine, and the shape of tool and shank will be treated in its proper place.

The Screw-cutting Lathe.—Plate V. shews various views of this, the oldest but most useful tool. The example is the design of the Britannia Company, and has 10 in. centres, that is, will accommodate work of 20 in. diameter (called in America a 20 in. lathe). 40 in. work can be turned by removing the gap bridge A, which is bolted down and dowelled, so as to allow the saddle to pass over it freely.

In all lathes the work is rotated, and the tool fixed in (usually) a *slide rest*, which can be moved along the lathe bed. This appliance, the very foundation of machine-tool accuracy, was the invention of Henry Maudslay. On account of the various diameters to be turned, the angular velocity must be capable of

variation, for the linear velocity at the surface of the work must be constant. Fig. 133 shews that if ab and $a_, b_,$ are equal, the angle $a_,''c\, b_,$ must be greater than $a c b$.

Let r = radius of work in inches.
V = speed of cut in feet per minute.
N = revolutions per minute to produce V.

Then, $\frac{2\pi r N}{12} = V$ and, $N = \frac{1{\cdot}91\,V}{r}$

And as the cutting speeds for roughing are, say:—

For brass	60 feet per min.
For gun metal	50 „ „
For cast iron	40 „ „
For wrought iron	40 „ „
For steel..........................	30 „ „

We have:—

Revolutions per m. for brass	=	115 ÷ rad. in ins.
„ „ gun metal ...	=	95 ÷ „
„ „ cast iron ...	=	76 ÷ „
„ „ wrought iron	=	76 ÷ „
„ „ steel	=	57 ÷ „

(*See also Appendix V., p.* 974.)

To effect this variation without altering the angular velocity of the main shaft, cone pulleys and back gear are employed.

The cone pulley C is driven by a belt, from a like pulley on the countershaft overhead, but the latter is reversed end for end, so that its small diameter is opposite the large diameter on the headstock. As the sum of driving and driven pulley diameters is constant, the belt will fit any pair, and a change of velocity will be effected, the highest being due to the smallest pulley on the head-stock. (See Fig. 539, p. 533.)

But as sufficient variation cannot thus be obtained we use the spur wheels known as back gear. The mandrel D (Figs. 131 and 134) is attached directly to the work by a driver. But the cone pulley runs loose upon the mandrel. Referring to Fig. 134, the bolt E serves to connect the pulley with the wheel F, which is keyed to D, and by sliding E radially outward till it engages between lugs G on the pulley, F and C are united, and the mandrel is driven *directly*.

Slower speeds are obtained by releasing E, and allowing the

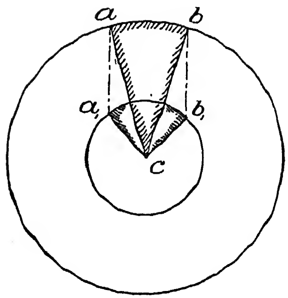
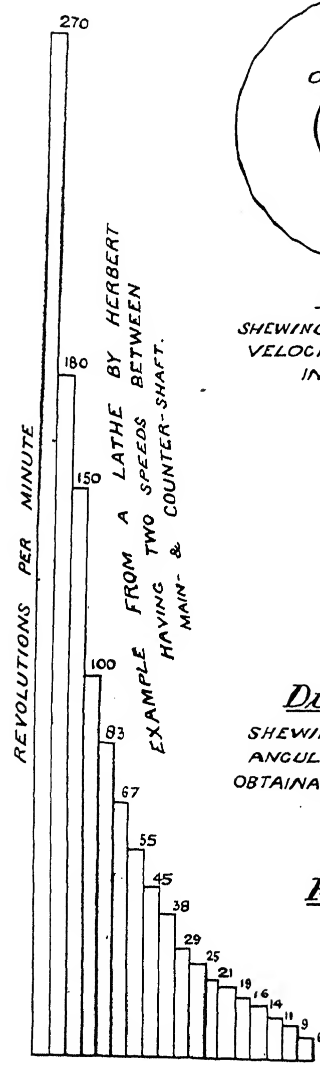
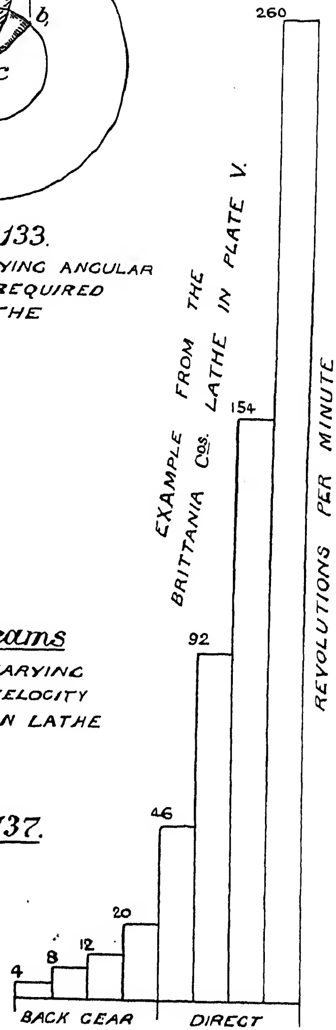


Fig. 133.

SHEWING VARYING ANGULAR VELOCITY REQUIRED IN LATHE





Diagrams

SHEWING VARYING ANGULAR VELOCITY OBTAINABLE IN LATHE

Fig. 137.

pulley to [illegible] only through the back gear [illegible] wheel [illegible] F, thus obtaining [illegible] inches on the [illegible] and B have each 45 teeth [illegible] Supposing the [illegible] have on machine [illegible]

[illegible]

With [illegible] gear [illegible] With back gear [illegible]

4 [illegible] 12 [illegible]

which is represented by diagram at Fig. [illegible] choose such pulleys as will give the correct cutting speed.

We have shown that when [illegible] the [illegible] pinion and wheel F are connected, at the same time J and K must be thrown out of gear. Both are keyed to the hollow shaft [illegible] revolving on the spindle M, which is supported on eccentric bearings [illegible] 'tommy' (or short rod used as a key) is inserted in the [illegible] and the spindle M is turned half round so as to throw the centres of J and K further from [illegible] and so disengage them [illegible]

The mandrel journals are conical, rotating in brass bushes (Fig. 134). Of these the left hand one is hollow, and [illegible] key, so that it may be tightened up after wear by screwing [illegible] nut [illegible] and the check nut [illegible]. The thrust of the work is taken on the end [illegible] when surfacing, and the head-stock is adjusted to secure parallelism by means of the screws in Figs. 133 and 136.

As the *fast* head-stock just described has no [illegible] position, the tail or *loose* head-stock must be adjustable for different lengths of work. It is shown in section at Fig. [illegible] and is fixed approximately by the bolt and clamp [illegible], after which the work is placed between the centres, and a fine adjustment given by rotating the hand wheel T, so pushing out the centre [illegible] (which acts as a nut with a left-hand screw), after which the handle V is used to clamp the barrel securely. Lateral adjustment of head-stock (when necessary) is given by the screw [illegible]

Fig. 139 gives the form of centre for supporting the work. It is shewn in position in Figs. 130 and 131, being securely placed in conical holes of fine taper, and has a flat surface for [illegible]

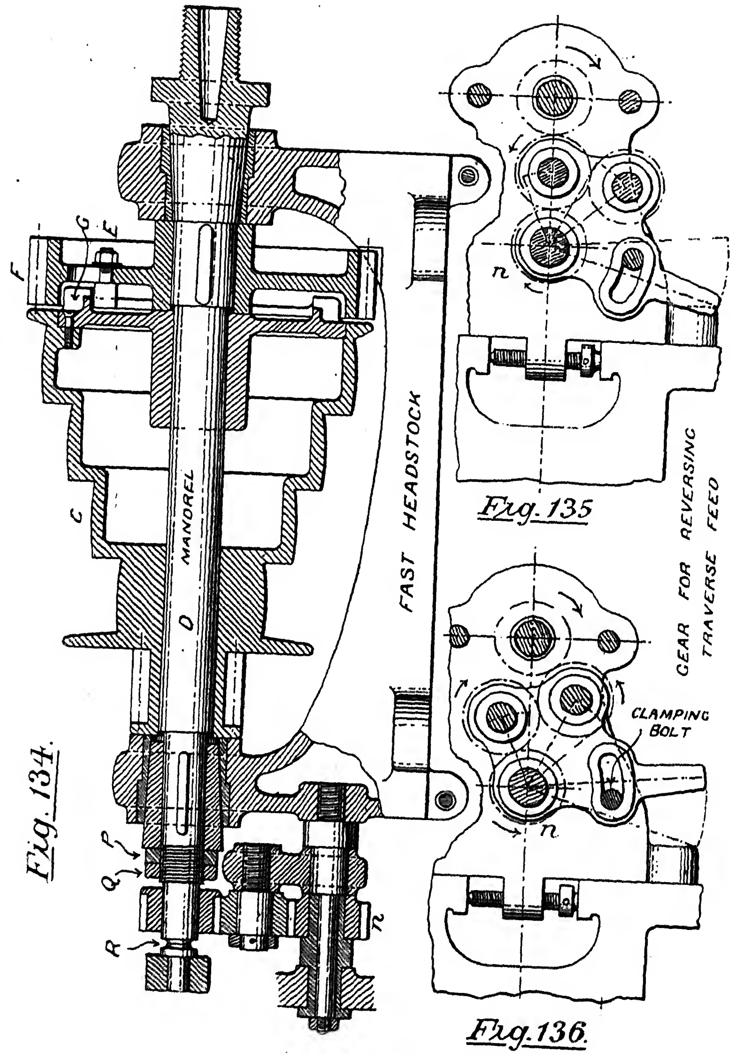
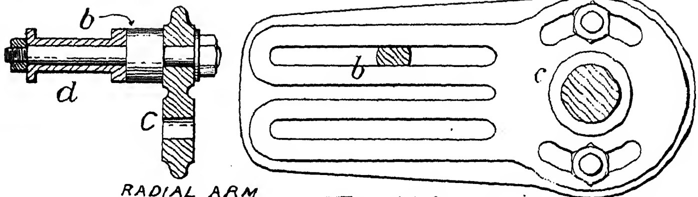


Fig. 134.

Fig. 135

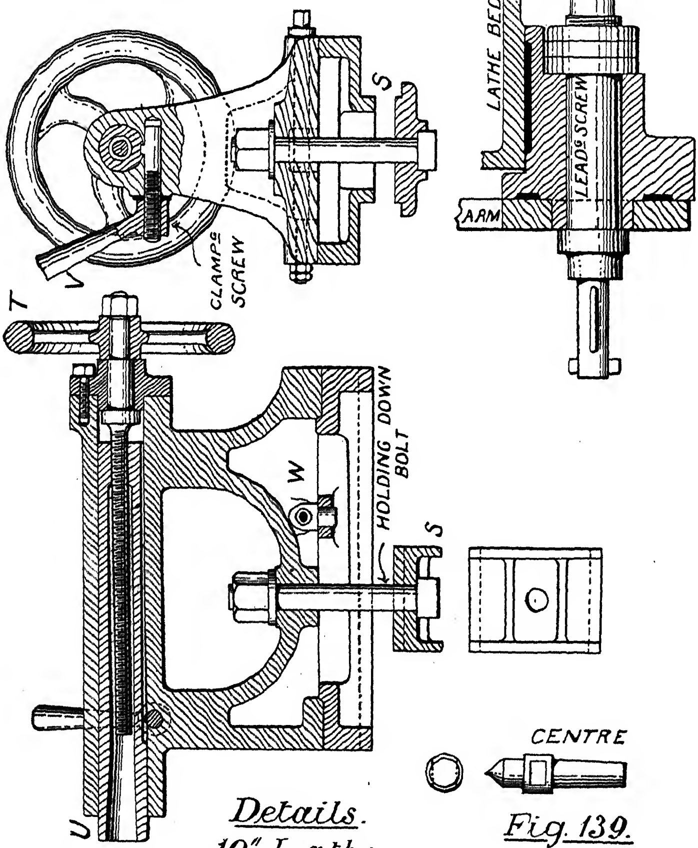
Fig. 136.

Details for 10" Lathe.



RADIAL ARM
FOR CHANGE WHEEL
STUD

Fig. 140



Details.
10" Lathe.

Fig. 139.

Turning to the *Slide-rest* and its various feed motions, details are shewn in Figs. 130, 131, 141, and 142. X is the saddle, having one movement, that *along* the bed; Y is the middle slide, moving *across* the bed; and the top slide Z has a universal motion, but by hand, being mounted on a circular table formed on Y; and thus a feed may be obtained at any angle by turning the upper plate Z and clamping the bolts *a a*.

The movement of X is called *traversing* or *sliding*, and the cross-movement of Y *surfacing;* these can be combined if required. The slide rest is actuated from the mandrel in two distinct ways. The *leading screw* at the front of the lathe bed is only used for screw cutting, and is thus preserved from wear at other times. It is driven by '*change wheels*,' at the left end of bed (Fig. 132). These can be changed, so that various rates of rotation of screw can be effected, relative to that of the mandrel, which comparison fixes the fineness of thread cut on the work. To facilitate the fixing of the wheels chosen, the intermediate stud *b* is supported (Figs. 130 and 140) on a radial arm or quadrant *c*, which can be clamped at various angles, the two wheels on *b* being fastened together by keying to a loose sleeve *d*. The saddle and leading screw are connected or disconnected by the two half nuts *e e* (shewn apart in Fig. 141), which are brought together by moving the handle downward along the dotted arc, when the studs *f f*, carrying the nuts, are brought nearer the centre by means of the curved grooves.

The slide rest is also worked from the back shaft *h* on the opposite side of the bed, and the two feeds for surfacing and sliding obtained. The shaft is driven from the mandrel by change wheels (shewn dotted at *g*, Fig. 131), the intermediates being carried on the arm *c*. Some makers drive by belt, which may slip if the machine is being overworked, but there is no doubt that wheels give a more definite feed. Passing to the connection of shaft with saddle we refer to Figs. 141 and 130. A worm *j*, having a feather key, slides along the back shaft, being drawn along by the saddle. The power passes through an intermediate worm pinion 2 to the wheel 3, which, being keyed on spindle *k*, crossing the bed, rotates pinion 4 on the front side. This pinion, gearing into wheel 5, turns the rack pinion 6, and the traverse is

Cross Section X A   *Fig. 111*   *10" Lathe*

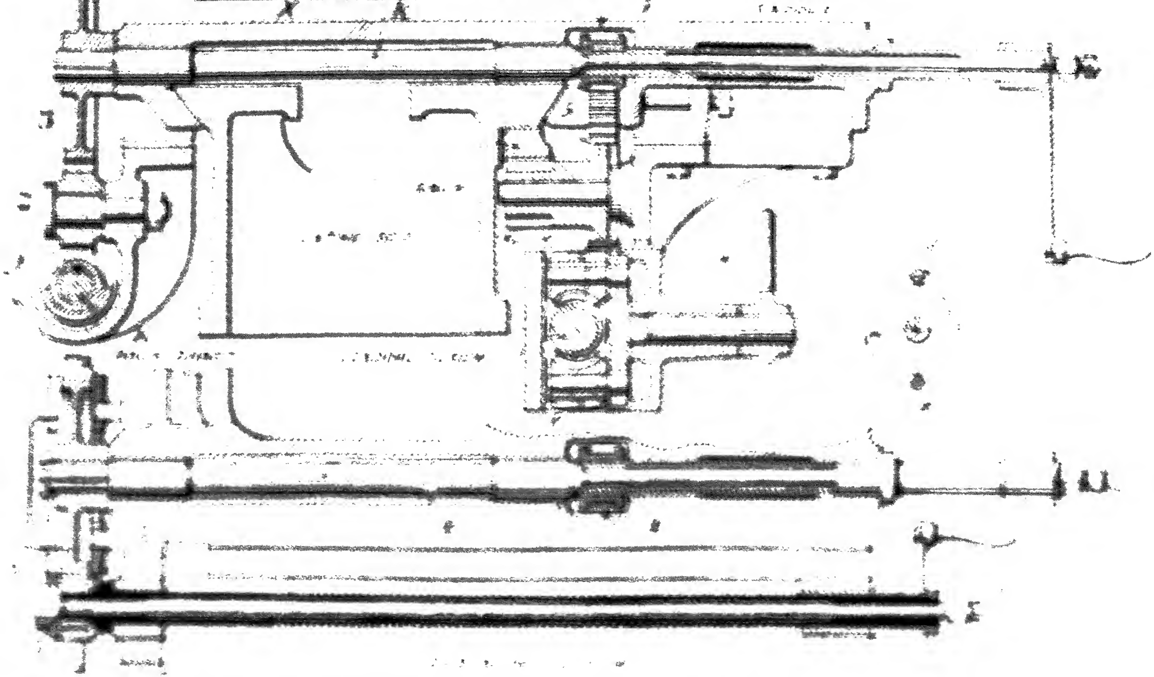
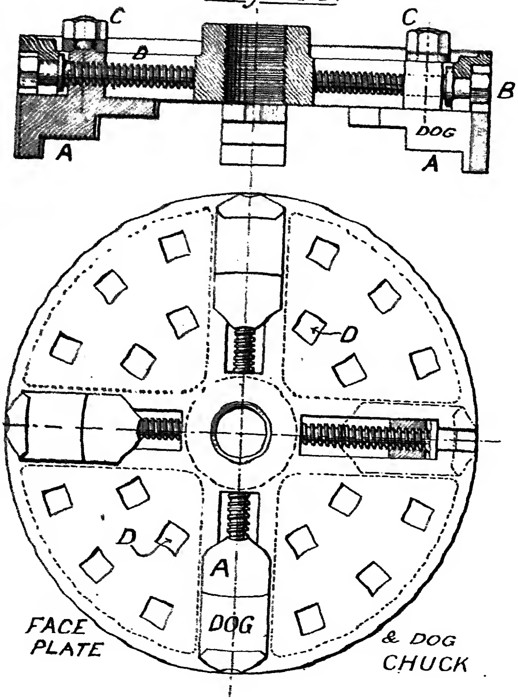


Fig. 143.



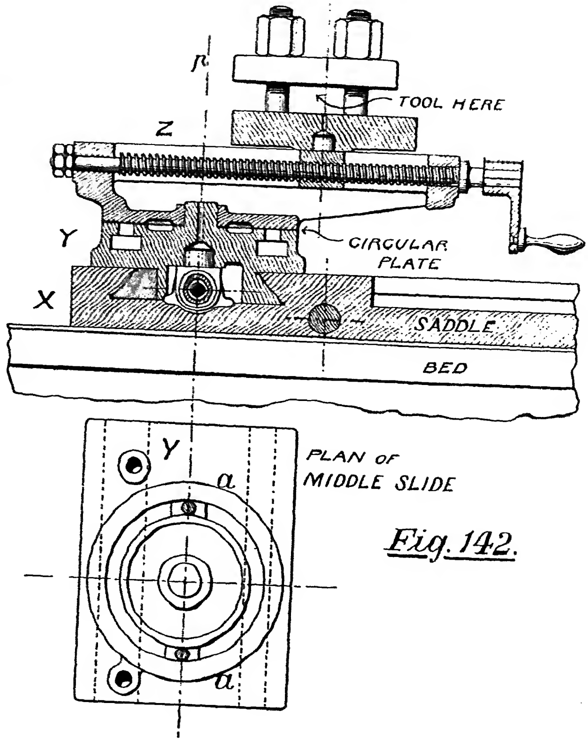


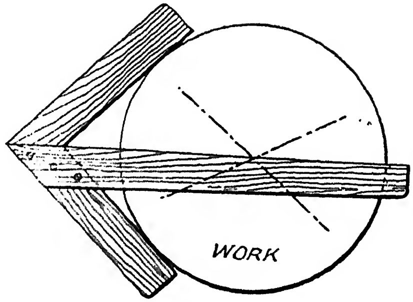
Fig. 142.

obtained. [illegible]
facing [illegible]
will [illegible]
The [illegible]
[illegible]

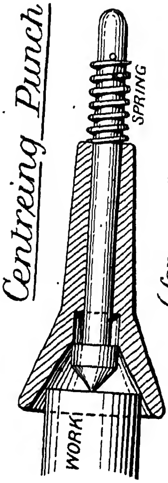
[illegible]
cutting [illegible]
required [illegible]
greater [illegible]
used [illegible]
turning [illegible]
handed rotation of the [illegible]
[illegible] changed [illegible]
of [illegible] obtained. [illegible]

Supporting the Work on the Lathe. [illegible] necessary to shew how the work is [illegible] spindle, it must be *centred*. Being [illegible] be first straightened [illegible] and the [illegible] is marked on the ends, either by *centre square* [illegible] drawing the dotted lines shewn [illegible] Fig. 145. The latter, being held [illegible] hammer. The centre is now formed, at least for the end [illegible] (the most important, because [illegible] stuff), it is next punched with [illegible] just support the bar in the lathe.* A [illegible] placed in the loose head-stock, [illegible] Fig. 139, but is sharpened on four sides only, instead of all round (see Fig. 146). Being hardened it serves as a tool for cutting the conical hole in the end of the work. Placing the bar between a conical centre in fast head-stock, and a square centre in loose head-stock, it is revolved carefully and marked with chalk where 'full' (that is, stands out more than the average), which if very bad may compel us to further straighten the bar. Then a [illegible] tool (Fig. 147) is placed in the rest against the bar, the latter being rotated rapidly, and the screw in the loose head-stock is turned so as to very *gradually* advance the square centre into the work. This centre hole must not be larger than necessary, and

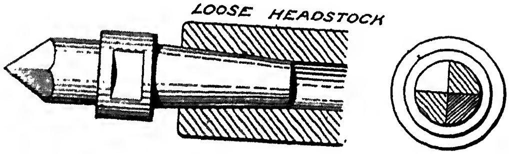
* See also [illegible] Fig. [illegible], p. [illegible]



Centreing Square
(for large diameters)
Fig. 144.



Centreing Punch
(for small work)
Fig. 145.



Square Centre Fig. 146.

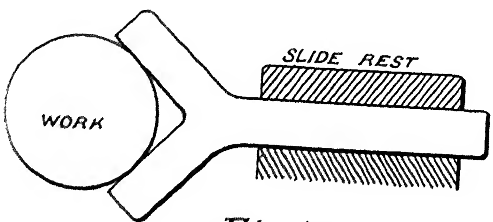


Fig. 147.
Crotch Tool.

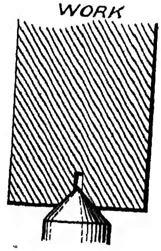


Fig. 148.
Finished Centre.

after [illegible] [illegible]
[illegible]

[illegible]

and the [illegible]

The [illegible] 60° [illegible] centre [illegible] is [illegible] *Appendices* [illegible]

Square centering [illegible] ever, proved satisfactory [illegible] work may be [illegible] *hand* [illegible] must always be [illegible] *centering machine* [illegible] been devised, [illegible] work [illegible] centred and [illegible] drills being alternately advanced [illegible] *Appendix* [illegible]

Driving. The above-mentioned [illegible] is driven from the mandrel as in Fig. 149. A carrier [illegible] the end of the work, and a catch plate [illegible] screwed on the mandrel, [illegible] projecting far enough to strike the carrier and rotate the work. The carrier shown will take work of varying diameter.

Other methods of support are the *Face Plate* and [illegible] *chucks* (see *Appendix* [illegible]).

The Face Plate, Fig. [illegible], [illegible] a large *dog chuck* having four jaws or dogs [illegible] independently [illegible] screwed on to mandrel, the fit being very accurate [illegible] correct *surfacing* across it. The jaws are adjusted [illegible] applied to the screws [illegible] until the work is centred and gripped, when the nuts [illegible] are tightened, thereby securing the screws.

The boss of a pulley may be turned [illegible] chuck [illegible] being held in the disc rim, and the [illegible] while irregular articles can be clamped [illegible] the jaws [illegible] dogs are removed, and bolts put through the [illegible] Such an arrangement would be that of a face plate proper.

Chucks.—Four examples of Whiton's chucks are shewn in Figs. 150, 151, 152, and 153. The *Independent Chuck* (Fig. 150) is really a dog chuck. The screws may be turned by a square key at A, so far as to release the jaws altogether, which, being reversed, as at B, serve to hold drills when boring stationary work, or to take a longer grip on rotating work. Fig. 151 is a good example of a concentric or '*universal*' *scroll chuck*. Applying a key to the bevel pinion C, the wheel D is rotated, carrying on its opposite surface what, on reference to front view, is seen to be a

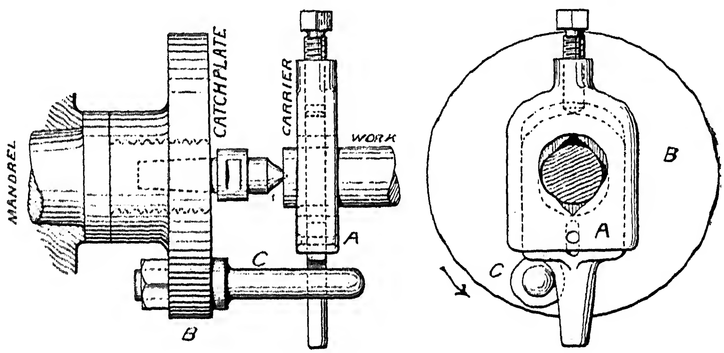


Fig. 149. Driving Arrangement.

spiral having three or four turns in its whole travel. The rotation of this 'scroll' moves the jaws nearer to or farther from the centre, *but equally*, thus centreing and gripping the work at the same time. Fig. 152 is a *Lever Chuck* having a scroll, but no gearing. A tommy is inserted at E to turn the scroll F, while the rest of the chuck G G is stationary. All these chucks are fastened to the mandrel in the same manner, by bolting to a small face plate screwed on the mandrel.

The *Drill Chuck* (Fig. 153) has the back portion H screwed on the mandrel, while the front part J carrying the jaws may be rotated; the scroll is therefore stationary while the jaws are carried round it. Hand tightening is sufficient for small drills, the surface of J being roughened for grip; greater tightness is obtained by using

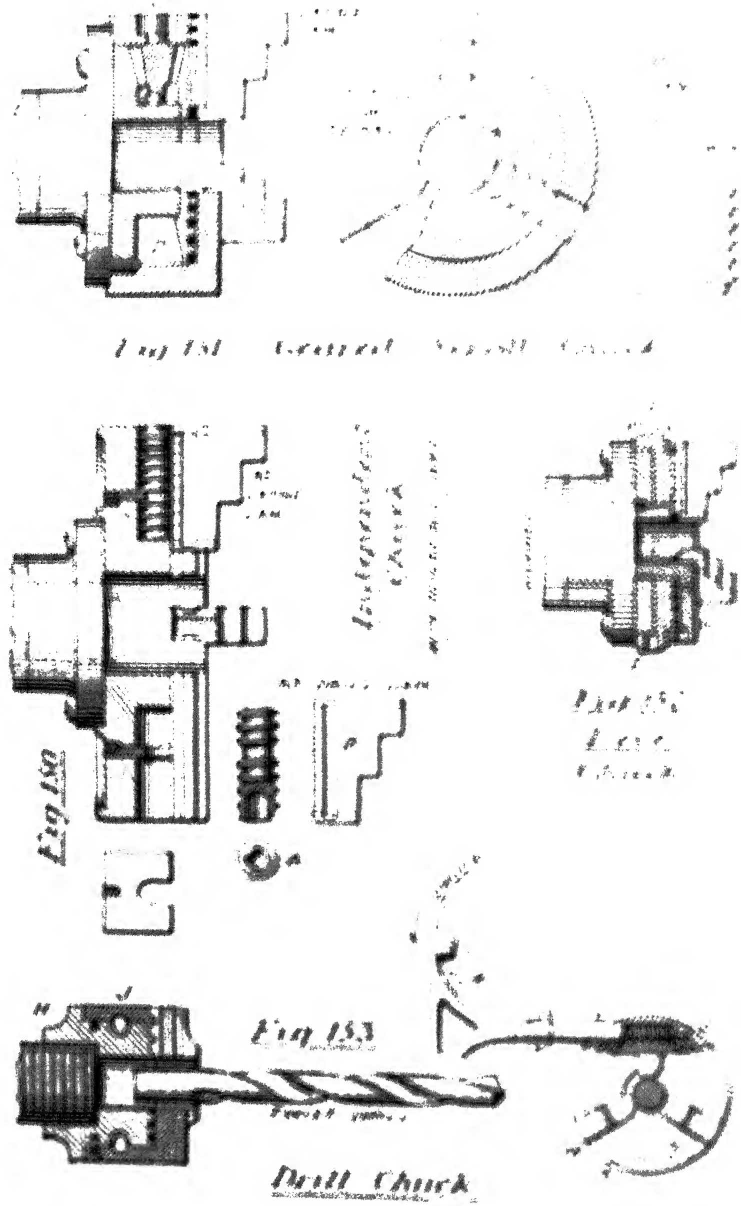


Fig 151 Central Scroll Chuck

Fig 150

Independent Chuck

Fig 152

Fig 153

Drill Chuck

the key as shewn at K; and, finally, the worm end of the spindle is used, as at L, for large drills. As the worm only bears on J in one direction, it is applied at the opposite hole M to *release* the drill.

Chucks that are either *independent*, *universal*, or *eccentric* at will, are also made, having combinations of the foregoing motions.

Expanding Mandrel.—There is still another plan of support for work having a hole through its centre. It is fixed on a mandrel (or spindle that can be centred in the lathe), of which several sizes are kept, having a slight taper, one suitable for the work being chosen; but a more expeditious tool is the *expanding* mandrel in Fig. 154. The mandrel proper is coned at A, and has three grooves of the same inclination as the cone, in which

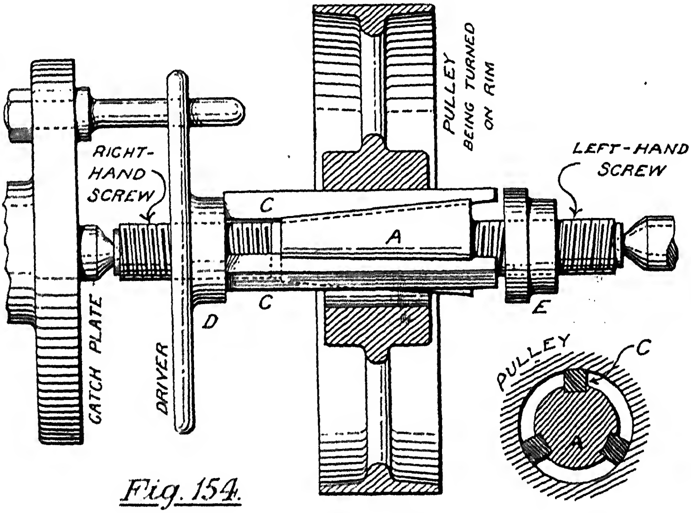


Fig. 154.

Noble's Expanding Mandrel.

ride keys so tapered that their outer surfaces form portions of one cylinder. The mandrel is screwed with right and left hand threads as shewn, and the advance of nut D will push the bars C C up the incline, so expanding the cylinder to any diameter within the limit of the tool. D serves also as carrier for the work,

and nut E on the right is for releasing the keys or for steady them. This tool is made by the Britannia Company.

Cutting Tools for Lathes.—There are various opini on the proper shapes of these. Fig. 155 shews the most comm where A is the plan of a straight tool, B that of a right hand t and C of a left hand tool; D being elevation for all three. Th

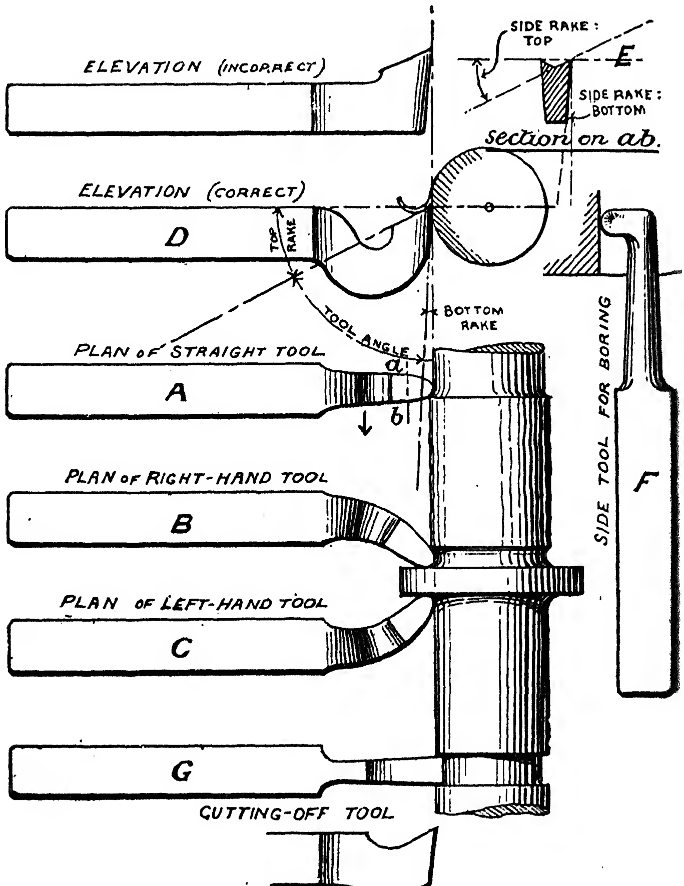


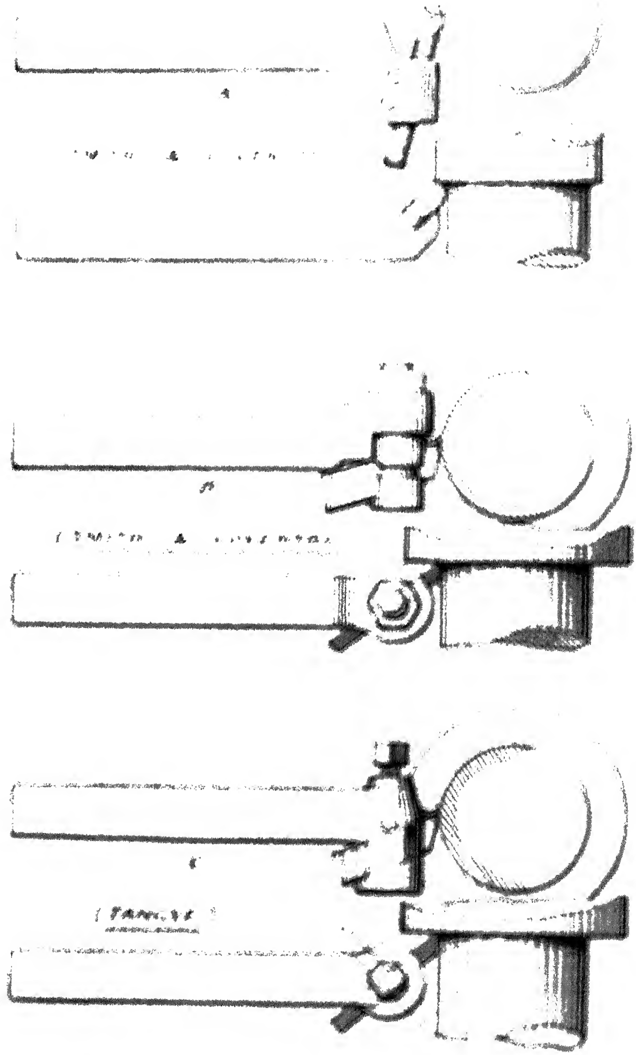
Fig. 155.

Cutting Tools for Lathes.

[illegible] of which each [illegible] and [illegible] form [illegible] cutting side, and tool angle, [illegible] cannot be given for all, for if the [illegible] at the side as well [illegible] shows the hollow cone required at top [illegible] in grinding, but may be approxi[illegible] side tool for boring work on the face plate, [illegible] a cutting-off or parting tool. These tools should all be set so that the point and the top surface of the tool are both at the level of the centre of the work. If set higher the tool spring will tend to lift it out of the cut, and if set lower it will give a tendency to [illegible]. The best method of clamping down the tool is seen in Figs. [illegible] and [illegible], Plate V., and in Fig. [illegible], where two [illegible] the tool by screwing down four [illegible]. Space is provided between the bolts to [illegible] adjustment. The shanks of cutting [illegible] dimensions, both for rigidity and for [illegible] cutting, otherwise the tool may [illegible] with a tool having a broad [illegible] considerably under the roughing [illegible] which cannot be remedied by a single [illegible] are taken before the final one, [illegible] *gradually reduced*. If the work [illegible] with length, a *back rest* is fixed on [illegible] grasps the work near the tool, [illegible] *Figs. I., II.* [illegible]

Tool Holders [illegible] used for some time, to do away [illegible] the tool when worn. These are [illegible] small pieces of steel, of [illegible]. They were first introduced [illegible]. Fig. [illegible] shows several tool [illegible] advantages [illegible] has the best form, though [illegible] was designed to obviate that difficulty, [illegible] though slightly different in detail.

The Break Lathe [illegible] because of a large 'break' [illegible] to admit a very large face plate [illegible] Plate [illegible] by Messrs. [illegible] & Bailey



Tool Holders Fig 156.

The fast head-stock B has a large cylindrical bearing at C, with adjustable cap, while the pressure of the surfacing cut is taken by the collars of the thrust bearing D. The face plate requires no further description than that given for Fig. 143, except to say that the jaw screws themselves take the grip, and that the jaw boxes may be unbolted and the work attached directly to the plate. The back of the plate has an annular spur wheel, driven by a system of 'treble gear.' We may turn the mandrel through the four wheels E F G H in simple back gear; or directly, bolting H to the cone pulley, and throwing out F and G by turning eccentric bushes; but if a slower speed be desired G is slid to the right, E and F kept in gear, while wheel K and pinion M, keyed to the third shaft L, are moved to engage respectively with pinion J and wheel N on face plate.

We have, therefore, three alternatives:—Direct driving without gear; double-purchase gear, E into F, and G into H; or treble-purchase gear, E into F, J into K, and M into N. The last is only required for large diameters of work.

The leading screw, lying within the lathe-bed at A, is driven, by change wheels P, through shaft Q, and wheels R R at the right end of bed. By removing the change wheels, the backshafts may be put in gear, the power being taken from the belt T, passing thence to the worm shaft U by spur wheels, and across to the rack pinion, as in the previous lathe. The handle V will pull the lever W, and clamp the leading screw nuts, while the traversing motion may be reversed at X. The slide rest has the same motions as have been described for Plate V., and the loose head-stock needs no further description.

This machine is used:—(1) As a screw-cutting lathe with or without gap; (2) as a *face lathe.* For the first the gap may be varied by loosening the bolts which hold the bed Y to the foundation Z; and by then applying a lever to boss K to turn a rack pinion *b b*, so bring the bed nearer the face plate, the standard *d* being also removed. The work would be supported between the lathe centres, and driven by a bolt in the face plate, or by small drivers as usual.

As a **Face Lathe**, the gap is widened; and the upper parts *e f g* of the slide rest being removed, they are bolted on the

standard at *h*, which has a circular T groove to receive clamping bolts, and admit of adjustment at various horizo: angles, thus obtaining a traversing, surfacing, or oblique fe The position of the standard is adjusted by loosening its four tion bolts, and applying a crowbar to the teeth *j j*. Feed is gi by hand, but can be made automatic as a *star feed*, or by overhead chain. By the former a star piece is keyed to the sl screw, and a projection on the face plate catches this at ev revolution, giving it a small turn. By the second, a ch attached to a crank pin on left end of mandrel, and passing alc overhead pullies, actuates a ratchet on the slide screw, and gi a small feed at each rotation.

If a face lathe be especially made for surfacing and very sh traversing, the bed is placed *across* the line of centres. (*Appendix II., p.* 808.)

The Boring Machine.—Figs. 161 and 162, Plate VI represent two views of a horizontal boring machine designed Messrs. Buckton & Co. As already mentioned, many bori machines are made with vertical bars, as for marine engi cylinders, the object being to balance the boring head, and p serve truth of surface; but if the bar be made very large a: rigid, as in the example, inaccuracy need not be feared. The are two classes of horizontal machine: in one the work is fix on a stationary bed, while the cutters travel, and in the other t bed and work are advanced, the cutter bar having no longitudir movement. The latter is analogous to lathe boring. (S *Appendix II., p.* 808.)

Referring to Figs. 130 and 131, Plate V., a cylindrical bar placed between the lathe centres, and driven by catch plat About half-way along this bar a longitudinal slot is made throu it, and a projecting cutter securely wedged therein. The upp slides Y and Z being removed, as well as the bearings *q* and Fig. 131 (made separate for the purpose), the work is bolted saddle X, by bolts placed in T grooves S S, and, as the bar rotat and gives the cut, the traversing feed advances the work to the toc

Boring machines are made on these principles, being, in fac lathes specially designed for boring. The bed is made low, an the fast head-stock high, the loose head-stock dispensed with, an

[illegible] arranged [illegible] A tube carrying the work [illegible] [illegible] to accommodate various depths of [illegible] *Fig.* [illegible] and *Fig.* [illegible]

[illegible] however, we have recourse to a machine [illegible] Plate VII, which is there driven by an engine [illegible] We will first describe it as though driven by [illegible] shaft, the usual plan. The worm shaft *s* would [illegible] with a [illegible] pulley to take the power from the [illegible] and [illegible] rates of rotation to the machine. [illegible] and worm wheel effect a slow rotation of the [illegible] which is the facing head [illegible], for surfacing the [illegible] bolted to the head [illegible], the tools [illegible] being [illegible] position. [illegible] catches the stop [illegible] after every [illegible] revolution, and gives a small turn to the feed screw. [illegible] the bar is the boring head [illegible], carrying alternately in [illegible] its circumference cutters or dummies, the latter to [illegible] the head in the cylinder and prevent 'chattering', this is [illegible] at Fig. [illegible] As the head revolves, it is fed slowly along the bar. Referring to Figs. [illegible] and [illegible] it is seen that the head [illegible] on its inner surface, and sliding in a groove in the bar [illegible] the screw [illegible] engages with the nut, and it follows [illegible] cause the head to advance, giving the feed. [illegible] rotation is obtained by spur gear at the right end of bar, where four wheels [illegible] form an epicyclic back gear, giving [illegible] slightly greater speed than [illegible], the difference of velocity being communicated to the screw through the pinions [illegible] and [illegible] When [illegible] and [illegible] may be slid out of gear by unscrewing [illegible], and the wheel [illegible] will then be used for hand feed or adjustment [illegible]

[illegible] describe the engine [illegible] is the cylinder, and *b* the crank shaft, [illegible] the steam entrance, and *d* the exhaust. A fly wheel on crank shaft carries the crank pin, and the motion passes to the worm through bevel pinion and wheel *f* and *g*, the remaining gear being as before. We must not omit the ingenious method of [illegible] the engine speed. The governor *h* is driven by a strap [illegible] on cone pulleys [illegible], having an ample number of steps. If we attempt to give the governor a high velocity the tendency is to throttle the steam and produce a lower speed on the crank,

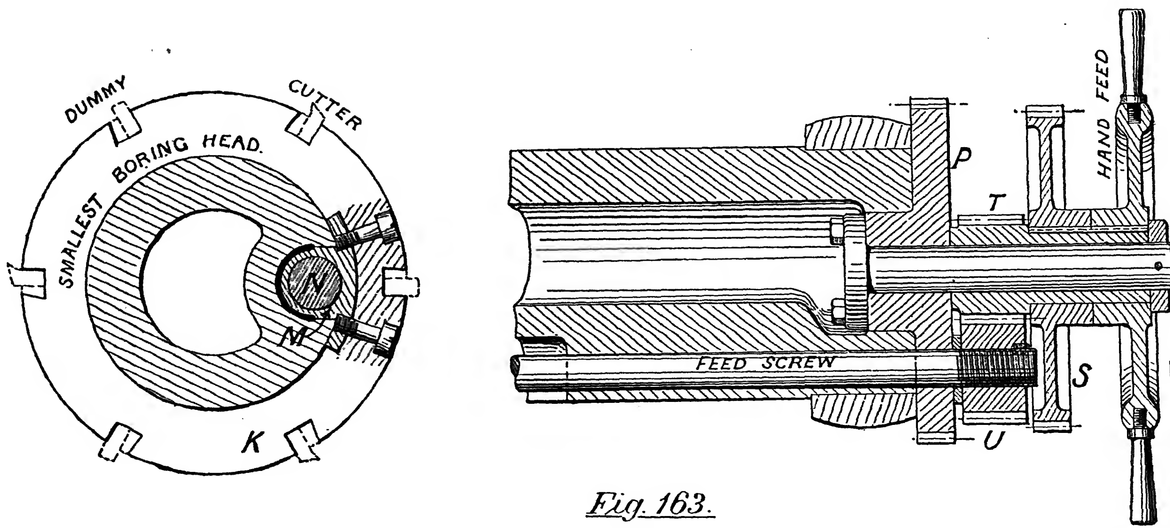


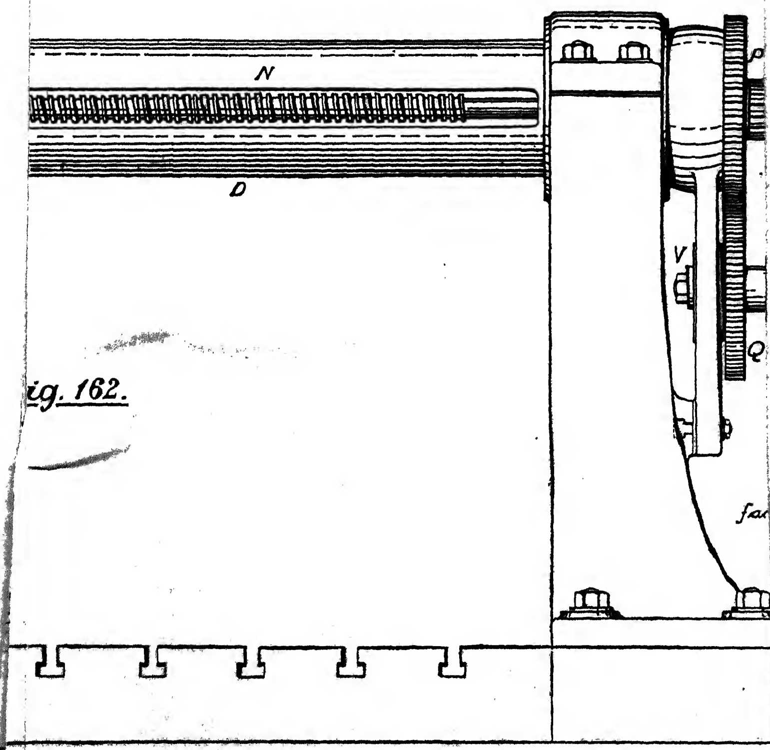
Fig. 163.

Sections of Boring Bar.

PLAT

& ENGINE COMBINED.

ı Buckton & Co.)



ig. 162.

[illegible], the effect of a decreased governor velocity is to admit [illegible] increase the crank speed.

The Drilling Machine. This, again, occurs under various [illegible] and single geared, radial, and multiple drills [illegible] in Chapter VII., being applied to [illegible] we will describe in order.

The Double Geared Drilling Machine is shown on [illegible] Messrs. Smith, Beacock, & Tannett [illegible] the lathe; thus the cone pulley A [illegible] wheel B, which is permanently [illegible] the bolt C and putting wheels [illegible], we may have a slower [illegible] of large diameter; E and F [illegible] shaft G in eccentric bushes [illegible] are transferred to [illegible] wheels [illegible] fixed to a sleeve K [illegible] at Fig. 165. The sleeve K carries a feather key M, fitting in a long key way in the drill spindle, thus allowing the latter to rise and fall. Passing upward, a smaller spindle N is attached by a pin, and forms one piece with M, while a loose sleeve P, having a rack formed on its right side, is held on the smaller spindle by nut O and check nut R. The rotation of a pinion S will thus raise or lower the drill spindle without affecting its rotation. Steel plates at T T diminish the wear caused by thrust of drill or weight of spindle.

The feed motion thus obtained is worked by hand or automatically. A worm wheel U, Fig. 164, on rack pinion shaft, is rotated by the worm on the spindle V, which takes its motion from the mandrel through another worm and worm wheel W, driven by cone pulleys X X, to give varying rates of feed. If we wish to feed by hand, use is made of the arrangement at Y, shewn in detail at Fig. 166. A race *a* is formed on the boss of the worm-wheel which drives the vertical spindle, and a small crank *a* serves to lift or lower the worm-wheel, to put it in or out of gear with the worm. The handle *b*, whose movement is limited by the groove *c*, is shewn holding the wheel *out* of gear, when wheel *d* is used for hand feed or adjustment.

Coming to the table *e*, carrying the work, it is fastened by set

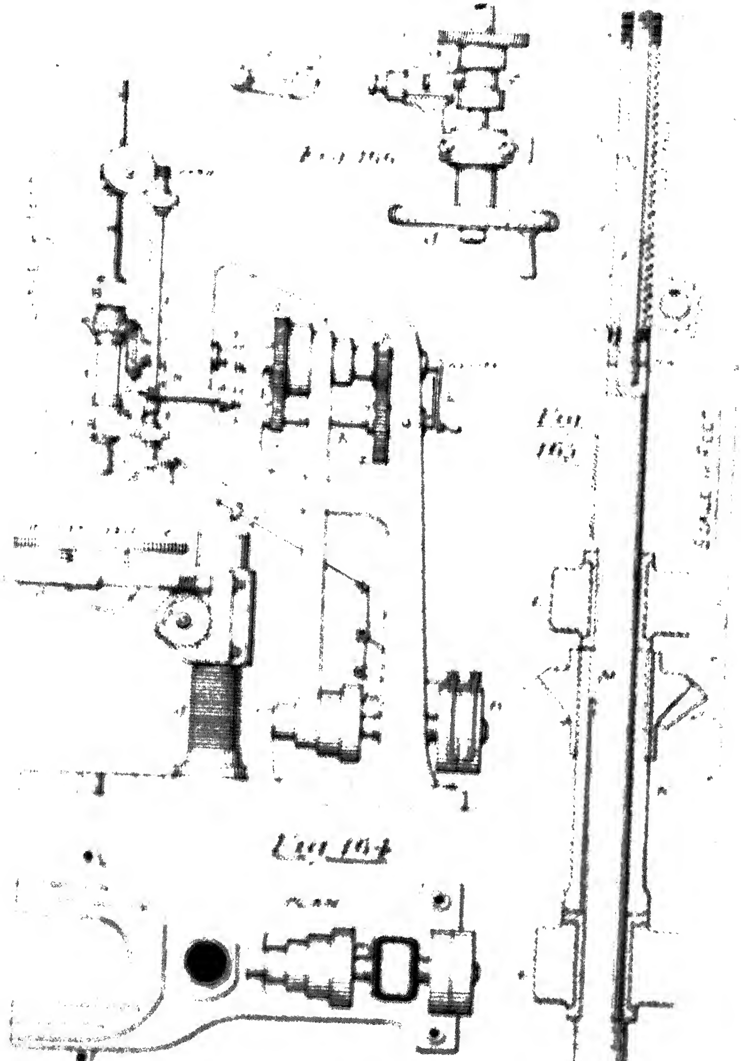
screw to a projecting arm *f*, and provided with slots for bolts, as in the lathe face plate. The pillar *g*, which supports *f*, has rack teeth turned upon it, so that the lifting apparatus may always remain in gear, whatever the position of arm *f*.

The lifting gear is as follows: A spindle *q* turns a worm, gearing into wheel *j*, which has on its axis a pinion engaging with teeth on the pillar *g*. The handle *k* serves either for spindle *q*, or for hand drilling when applied to the mandrel. Some machines have a plain pillar, as in the next example. A very deep piece of work is accommodated by bolting to the foot or bed, and swinging the table out of the way.

In double-geared drills the countershaft is usually self-contained, as at *m*; and the pulley *n* is driven from main shaft; the fast and loose pullies lying side by side, and the fork being moved by handle *p*.

The Single-Geared Drilling Machine in Fig. 167 needs little further description. Back gear is dispensed with, and the cone pulley A keyed to the mandrel. Hand drilling is provided for by the handle B on fly-wheel C. S is the hollow sleeve driven by mitre wheels; and a feed screw at D takes the place of the rack, being provided with a long key-way, while a key E (shewn black) is fixed to spur wheel F, so that a feed may be obtained at any height of drill spindle. The feed screw further passes through a nut G, fixed to the casting H, and a rotation of F will therefore raise or lower the screw; such rotation being effected by turning the hand-wheel on spindle J, the latter carrying a pinion K gearing into F. A socket L in drill spindle receives a cylindrical projection on the screw, in which a race is turned; and a pin M, passing through the spindle tangential to the race, allows the screw to lift the spindle without affecting the rotation of the latter. In the best machines the feed screw is a hollow sleeve.

The table and supporting arm are similar to the last example, the lifting gear consisting of a handle N and worm P, worm-wheel Q, and rack pinion, the rotation of the last lifting or lowering the arm. The rack R is a sort of strut fitted between the top and bottom collars of the pillar, but otherwise loose. If the table be moved horizontally the rack is carried round the pillar, and remains in gear with the pinion in all positions.



DOUBLE-GEARED DRILLING MACHINE

(Smith, Beacock & Tannett.)

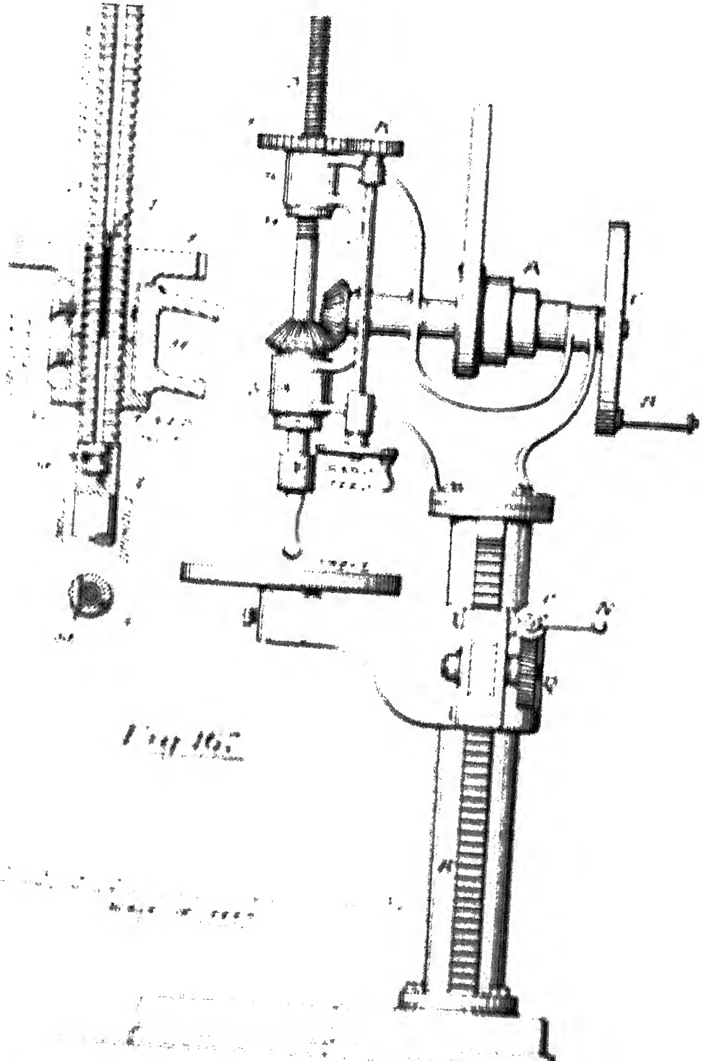


Fig 165

Single geared Drilling Machine

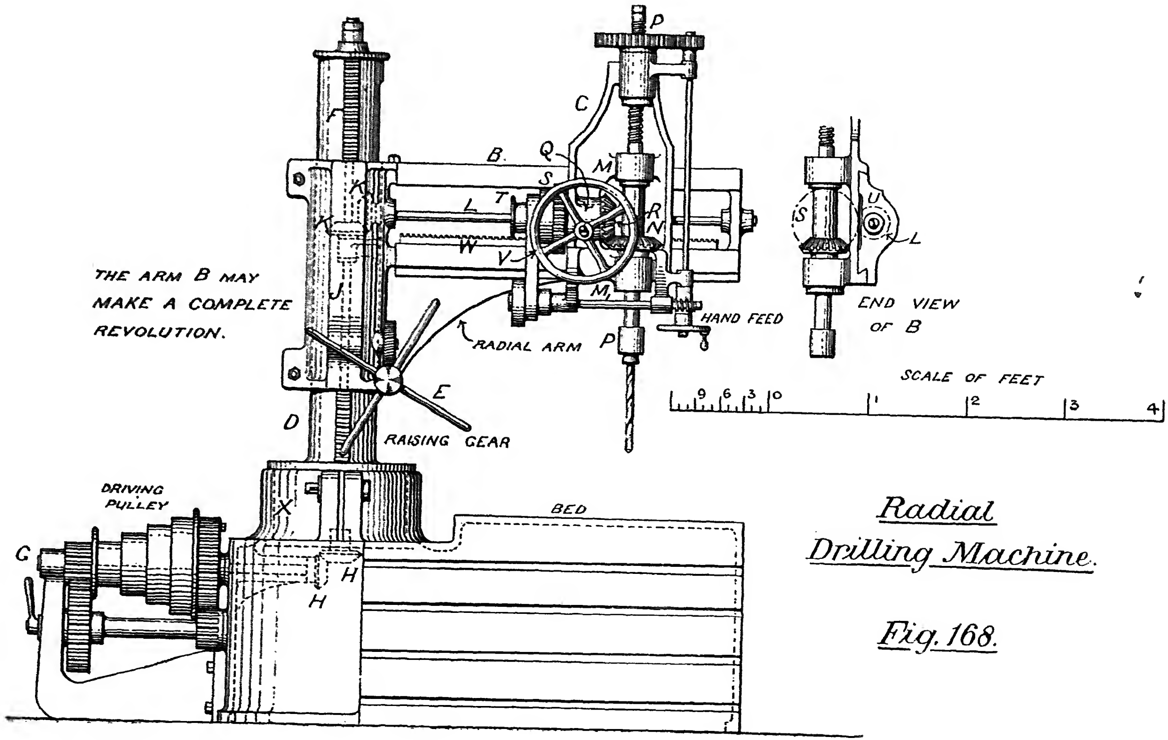
The Radial Drilling Machine is most useful fo work not readily moved, and has been, since first designed in request for holes in steam cylinders or boilers. The form depends on the nature of the work, and is sometimes dis with, and a trolley run under the drill. Then the radial ar be swung from a wall or roof stanchion. The machine in F has a stationary table, to the top or side of which the w bolted; and the tool is adjusted over the work (1) by an a movement of arm B; (2) a traverse of the saddle C; (3) a fall of B. The last is obtained by turning the spokes E, through worm and wheel, rotate a pinion in rack F; and first both arm and pillar turn within the bed at X. The m G is driven directly or by back gear, and mitre wheels H H tr its motion to the spindle J, from which again the power is to the horizontal spindle L by mitre wheels K K. As arm rise or fall, K is supported by a bearing projecting through in the hollow pillar D, and a feather key connects K and J.

The saddle C has bearings M M, and a sleeved mitre w drives the drill spindle P. A bearing Q supports a short s to which are keyed the mitre wheel R, spur wheel S, and pulley T, from the last of which various rates of feed are ob as usual; and power is given from L to S by a pinion U, w having a feather key, follows the saddle, so as to keep alw gear.

The drill power passes therefore through five shafts, G, J, and P, but this is not considered complicated in view o advantages obtained. The saddle is moved along the ar turning the hand wheel V, which rotates a small pinion, gearin the rack W. (*See also Chap. VII., p.* 303 *; and p.* 1018.)

Drills.—Some forms are shown at Fig. 169, where A i *pointed* and fits in taper hole in the spindle, the cotter *a* prev slip. The method of sharpening is seen at *b*, *c*, and *d*, and n *e e* increase endurance of point. B is a *pin-drill*, where var in diameter of circle cut is permitted by the movable cu wedged in the slot *g*, a hole being first drilled in the wor receive pin *h*. Cutting angles have been previously discusse

The *twist drill* C, no doubt the very best form for acc work, is much in favour. A socket *i* fits the spindle, and



Radial Drilling Machine.

Fig. 168.

the larger drills; a second socket *k* within the first is for medium drills; and a third, within *k*, fits the smaller sizes. These are carefully ground to fine taper, and are quite rigid.

The Slot-Drilling Machine (now metamorphosed into the vertical milling machine) has a saddle carrying the drill spindle

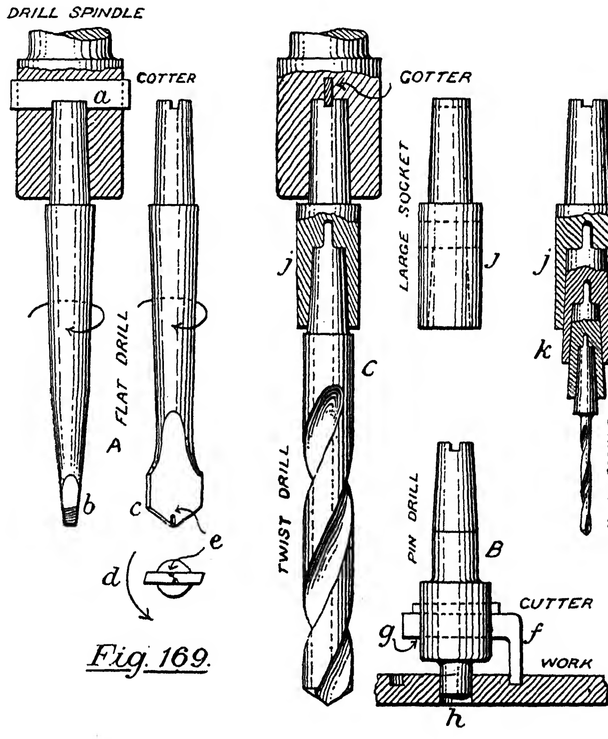


Fig. 169.

as in Fig. 168, but arm B is made immovable. While rotating, the spindle also receives a traverse along slide B, taken from a leading screw, lying within B, in addition to the shaft L. The drill thus cuts out a circle that travels along a straight line, known as a *slot.* Keyways and cotter holes are examples, and for such work vertical and horizontal feeds are required. (*See Appendix II., p.* 810.)

The Planing Machine, as mentioned, is not strictly economical, because the tool cuts in one direction only, and the back stroke is wasted. To minimise this loss, and at the same time reverse the stroke without changing the continuous rotation of main shaft, ingenious motions called *quick returns* have been devised.

A large-sized planing machine is given in Plate IX., as made by Messrs. Hulse & Co. The table, stiffened with ribs, and having **T** grooves on its surface to receive clamping bolts, slides in **V** grooves B B, made true and level, being the copies. Thus the work travels, and the tool is fixed. The belt pulleys C, D, E are loose on their shaft, but C and E are technically 'fast' pullies, because they drive the table, being fixed to pinions F and G. The strap being on pulley C, pinion F engages with wheel H; and pinion J on the axis of H gears with K; L in turn with M; lastly pinion N moves the rack P fastened to the table. A slow cutting advance is thus obtained. At the end of the stroke the strap is moved from C to E, and then K is driven directly from G, the rest being as before. Dispensing with one pair of wheels we have effected two objects—(1) a reversal of the stroke; (2) a quicker rotation of N, or *quick return* to the table. (*See App. III., p.* 918.)

When at rest the strap is on loose pulley D, and handle Q lies at right angles to the bed. Being connected to strap fork through levers R, S, U, inspection shews that Q moved to the right will give the advance, and a reverse movement the return stroke. But, once started, these motions are automatic, thus—Let the table be returning leftward in Fig. 171, back stop X will at end of stroke catch lever Y, and move it to the left, shifting the strap rapidly from E to C, the advance pulley. If the table travel to the right, stop Z catches Y and puts the quick return in action. These stops may be adjusted to give various lengths of stroke.

Two vertical standards *a a* bolted to the bed have slides on their front edges, and are stayed by tube *b*. A cross slide *c* lies across them, supported by vertical screws *d d*, passing through long nuts at the back. On the slide are two saddles *e e*, carrying other slides *f f*, to give a vertical movement to the tool. Screws *d d* are to adjust the cross slide to any desired height, after which it is clamped by screws *g g*. A handle may turn shaft *h*, which is

[illegible]

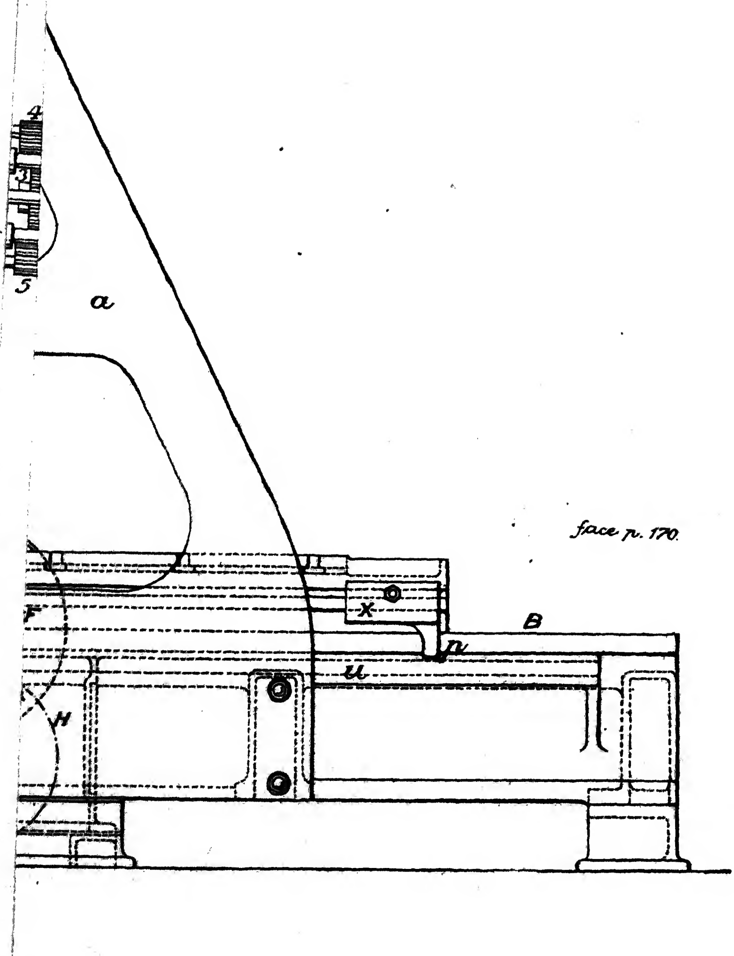
[illegible] the ratchet teeth. The pawl [illegible] motion is [illegible] returns [illegible] its first position [illegible].

The principal feeds [illegible] each with one of the saddles [illegible] and wheels [illegible] the right connect [illegible] and [illegible] with wheel [illegible] takes its motion from [illegible] gear [illegible] given in one direction [illegible] and [illegible] are [illegible] with them the direction of feed is reversed [illegible] mediate position [illegible] at rest.

A vertical feed is obtained when wheels [illegible] motion being given to shaft [illegible], and from [illegible] gear and horizontal spindle to the screws [illegible] screws are also for fine adjustment of tools. The boxes [illegible] T groove formed in the saddle, and [illegible] clamping of front slides [illegible], when the [illegible] feed becomes angular. Using suitable wheels, one tool may be fed horizontally and the other vertically or at an angle.

The tool box may also be fixed at a small angle, [illegible] the slot [illegible], and the front or flap [illegible] hinged at [illegible] that [illegible] may lift during the return stroke and avoid useless wear of [illegible] an automatic motion being sometimes provided for relieving [illegible]

PLATE IX.

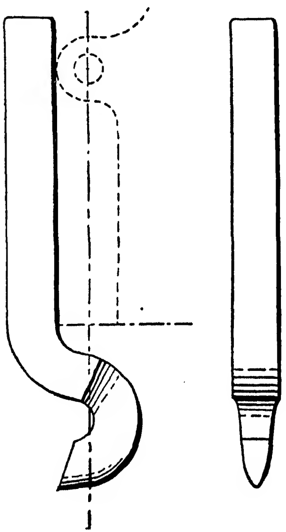


face p. 170.

tool altogether. Special clamps 23 hold the tool, having large square holes to receive it, and the turning of the set screw serves both to fix tool to clamp and clamp in tool box.

The tool itself is shaped as in Fig. 172, being so bent back as to place the point nearly in a line with the hinge and prevent 'digging.'

The Shaping Machine is a planer with moving tool and fixed work, having on this account some advantage for small articles; for if a moving table be employed, its stroke must exceed the length of work, so as to leave space for the acquisition of velocity in such a heavy mass; while the moving parts in the shaping machine, being much lighter, enable us to adjust the stroke with nicety, besides absorbing less work by friction.



Planing Tool.

Fig. 172.

The machine in Plate X. is by Messrs. Smith & Coventry. The tool box A is fixed to a 'ram' B, the sliding of which in saddle C gives the cut. The saddle moves along the bed D to give the feed, and an arm E, cast upon it, supports a rocking lever F, which actuates the ram through the rod H. The cone pulley rotates right-handed, carrying on its shaft (which extends the whole length of bed) a pinion K, giving wheel L a left-handed rotation. L turns on a stud fixed to the arm E, and carries a crank pin P, whose throw may be adjusted similarly to that in Plate XI. A die on this pin slides in a slot M, formed in the oscillating lever F. Referring to Fig. 174*b*, the *uniform* rotation of L will give the ram a slow advance when travelling from *a* to *b*, and a quick return from *b* to *a*, because *ab* is a longer path than *ba*, as shewn by the arrows; the proportion being 23 to 14

in the example. The pinion K can slide on shaft Z, and so keeps always in gear with L, being driven by a feather key. Length of stroke is adjusted by the position of P, but *position* of ram is given by adjustment of the nut Q.

The table R, supporting the work (which is bolted to top or side as found convenient), may be adjusted for height by the handle S, which, by mitre gear T, rotates the screw within the nut U, fixed to the bracket V. The horizontal position of the work may be varied by moving V along the bed to the point required.

W is a mandrel upon which hollow cylindrical work may be placed by removing the loose collar X, and gripping the work between the cones. The bracket Y steadies the end of the mandrel.

Three feeds are required, each of which may be worked by hand if desired. The pinion 4 (Fig. 174*a*) drives wheel *g*, carried on a stud *e*. An adjustable crank pin on *g* is connected to the lever *h*, which gives, through ratchet *d*, an intermittent rotation to the spindle *f*. Upon this spindle is a worm gearing into a worm wheel on the mandrel W, and thus a rotary feed is conveyed to the mandrel. The latter may be used for such articles as lever bosses, which are interrupted on one side by the lever arm, and therefore unsuitable for lathe work. The second feed is a horizontal motion of the saddle for work fixed on the table. A crank pin *k*, on the wheel L, is connected to the ratchet *m*, and the motion transmitted by *n* to the wheel *p*. *p* forms a nut attached to the saddle, and as the screw *q* is fixed to the bed, it is evident that a rotation of *p* will advance the saddle along the screw.

The third feed is vertical. *r* is a bracket fixed to the saddle, and *s* a rod sliding in *r*, as well as in brackets *t t* carried on the ram. At each back stroke of the latter the tappet *w*, on rod *s*, is caught by the bracket *r*, and *s* is moved to the left, causing the ratchet gear *x* to turn the screw *y*, and give a small vertical advance to the tool box. When the ram reaches the end of the advance stroke the tappet *z* in turn catches *r*, moving *s* back to its original position. The head A can be fixed at an angle to the vertical by unclamping bolts 2 2, and refixing, when the last-mentioned feed becomes angular; and the position of the tappets may also be varied.

In addition to the above, a fourth movement, enabling us to fix the tool box at an angle while preserving the vertical feed, is

PLATE X.

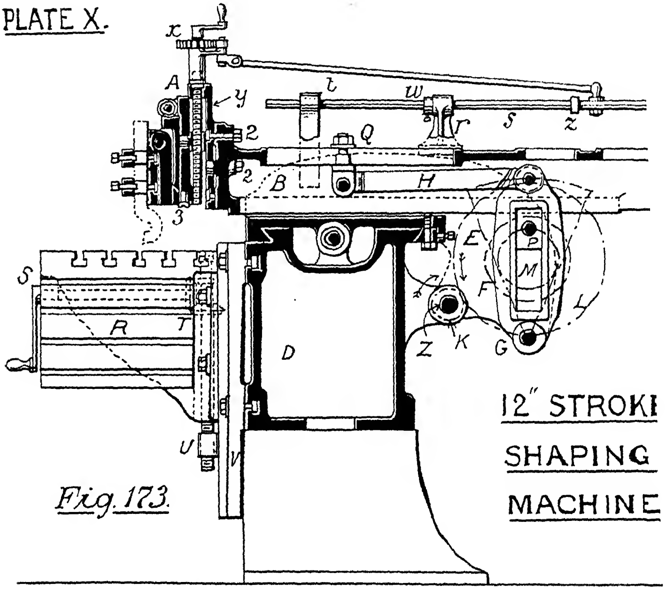


Fig. 173.

12" STROKE SHAPING MACHINE

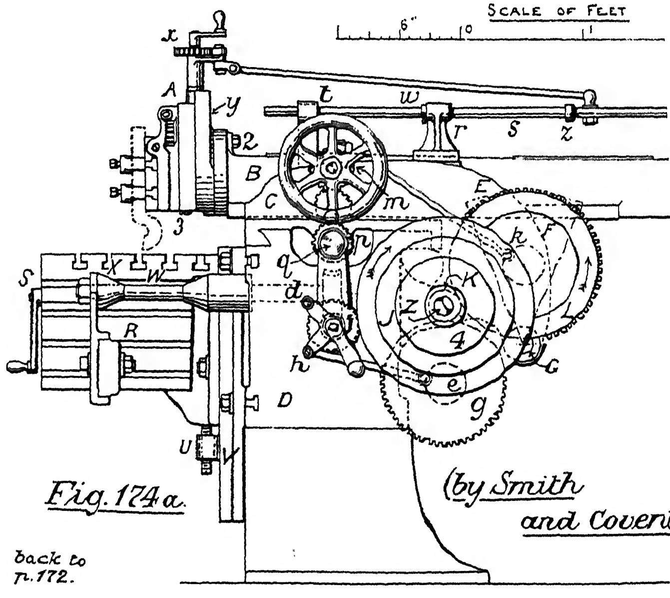


Fig. 174 a.

(by Smith and Covent

back to p. 172.

PLATE X.

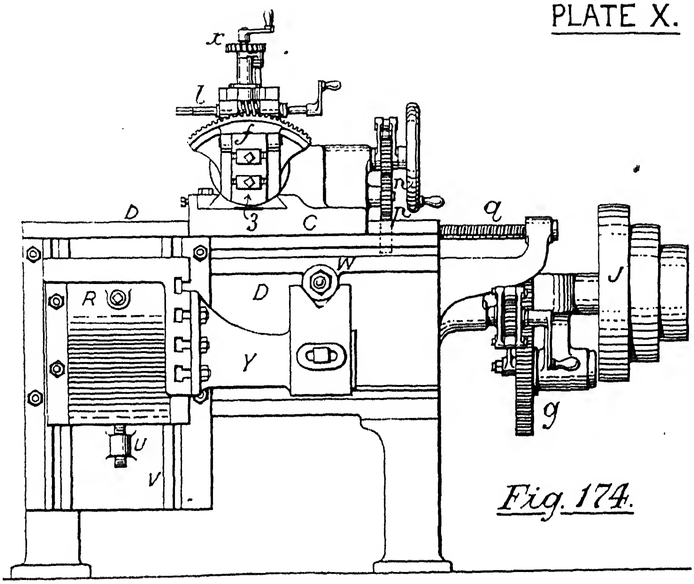


Fig. 174.

DIAGRAM OF QUICK RETURN MOTION

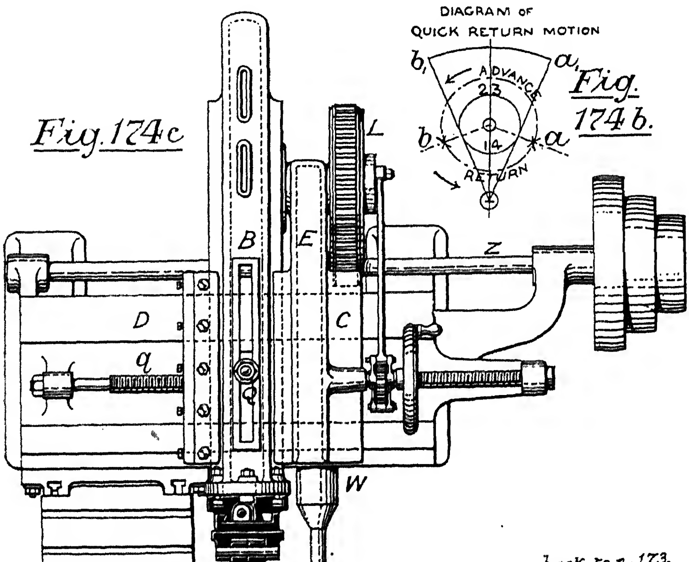


Fig. 174 b.

Fig. 174 c

back to p. 173.

obtained by means of the worm spindle *l*, provided with a handle, and worm gearing into the segment *f*, which is pivoted at 3. We may thus shape a corner or give a feed (by hand) for a concave surface. The front of tool box is provided with the usual flap to relieve the tool during the return stroke, and the tool itself takes the same shape as that described for the planing machine.

The Slotting Machine is probably the least economical of machine tools. While the planing machine takes simple horizontal cuts, and the shaping machine tools cylindrical work lying horizontally, the slotting machine is for the production of *vertical* cylindrical and plane services. Though working at a disadvantage in having to lift a heavy ram, this machine has served a purpose, and is still used to a large extent. Smaller work can generally be accommodated in a shaping machine, but the slotting machine is used for heavier work, and is made more powerful.

Plate XI. represents one of these machines, as made by Sir J. Whitworth & Co. Power being given to the cone A, it may be passed directly to the mandrel B, or through the back gear at C, the back shaft being moved to the left or right, or (Fig. 175) to put the wheels in or out of gear respectively. The power is further taken from the mandrel to the ram through the medium of a quick return motion. Looking at the front of the ram, and keeping our attention on both views, the spur wheel E is driven by the pinion F, and the motion transmitted to the crank disc G by pin H. The spur wheel turns on the boss J, and the crank disc in K, their centres being $1\frac{1}{2}$ inches apart horizontally. Referring to Fig. 177, if the spur wheel rotate uniformly it will pass through 10 divisions while bringing the pin from H to H_1, but through only 7 divisions from H_1 to H, and the advance will bear the proportion of 10 to the return 7. As some sliding takes place between pin H and disc G, a die is provided. The rod L connects the crank disc with the ram M, and there are two adjustments; one at N to fix the height of the ram; the other at P, where the rotation of two screws is made to move the pin and regulate the throw of the crank. A brake block Q, bearing on the crank disc, may be tightened by screwing up the wedge R, and serves to fix the ram in positions where it might fall on account of its weight.

There are three feed motions, all taken from cam S, the

Figs. 175 and 178, which is connected by a shaft with the di: At every rotation of the cam a vibration is given to the lev which is connected to the lever U (Fig. 179), carrying a rat pawl, and a partial rotation of shaft V (Fig. 175) thus obtai Both levers are provided with slots to adjust the amount of fe

The table W to support the work, is circular in form, and worm teeth on its lower rim. It is mounted on two slides X Y, which are again supported on the bed slide Z. The sha turns the bed screw *g* through the wheels *e* and *f*, giving a lo tudinal feed, useful for cotter holes and such like. Putting *f* of gear by sliding, a cross feed is effected by wheels *a* and *d*, former taking its motion from V by mitre gear, and the la being fixed on the cross slide screw *h*, so that Y would stationary and X would traverse. The third feed is a rotatio the table obtained by the worm gearing above mentioned; wheel *d* being slid out of gear, and *b* put in, the worm shaft rotated, and its motion transmitted to wheel *k*, cast on the ta This motion is analogous to that of the shaping machine mand

It has been customary to attach the tool directly to the ra and let the point scrape on the work during its return, giv useless friction and wear, but it is now recognised that a flap advisable, and such a tool box has been shewn. A spring on front or counter-balance at the back is necessary to bring the t back to its work, gravity not being otherwise employable.

The form of tool may be as for previous machines.

The Milling Machine, though in its present form recent introduction, has been known for a very long period; it was not till milling cutters or 'mills' were produced m cheaply and correctly by emery grinders that the principle co be sufficiently extended.

Cutter.—As already mentioned, a rotating cutter is employ to which the work is fed, and this we shall first discuss. Fig. 1 represents a spiral mill for tooling flat surfaces. All these mi are keyed to a mandrel or cutter spindle, which is either rotat between centres, or fixed into the catch plate and only centred its opposite end. Fig. 182 shews a key-seating or groovi cutter for cutting key ways or as a parting tool. Being grou both on circumference and sides, it becomes narrower at ea re-grinding, and therefore inaccurate. This can be avoided

PLATE XI.

ACHINE.

itworth & Co.)

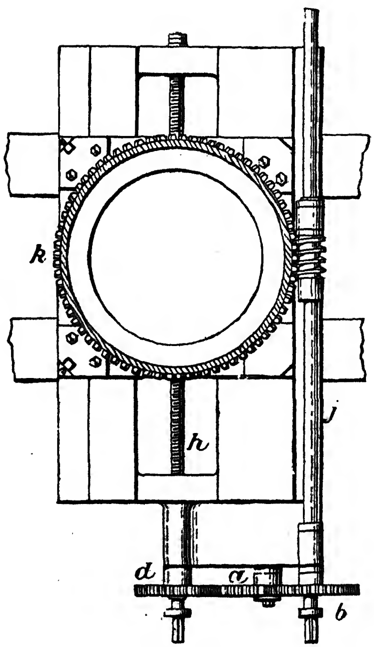
D
M

Fig. 178.

79.

Fig. 180.

PLAN OF SLIDES.



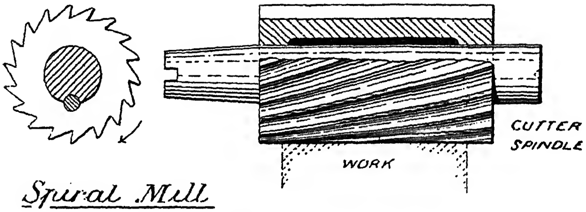
face p 174

the use of the expansible cutter in Fig. 183, which is divided at *a b* by a plane slightly inclined to that of the cutter, and has thin discs inserted to preserve the normal width. If *a b* were at right angles to the axis a strip of uncut material would be left on the work, which is here obviated, besides which, various widths of grooves may be cut. Further, if required, two mills may be placed on one spindle, the teeth being interlocked, and a groove of about twice the former width thereby cut, but it is important that the mills be of exactly the same diameter, obtained by grinding them together on the same spindle. Fig. 184 shews a pair of heading or twin mills for forming the sides of hexagon nuts or other parallel work, the width being varied by the insertion of suitable packing. In Fig. 185, A is a mill for grooving a screw tap, B for fluting a rimer, and C an angular mill for cutting the teeth of other mills. (*See Appendix II., p.* 811.)

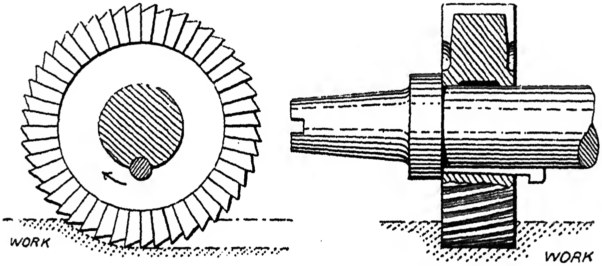
When a grooving mill is allowed to cut on its side only, say, when fixed in a vertical machine, it is termed a face cutter, but such an application is not desirable.

The steel or 'blank' to form the cutter is turned to correct diameter while soft, and the teeth then cut. It is next tempered to a straw colour, and the edges are finished by grinding with a small emery wheel of the same shape as the mill C, Fig. 185. Great care must be taken to avoid cracking while hardening, but distortion is now removed by grinding the *hardened* mill.

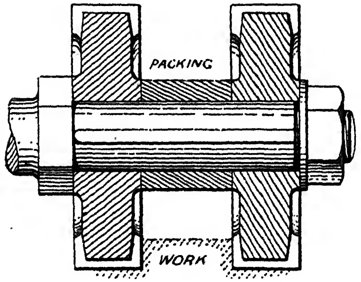
Fig. 186 represents a cutter for forming the teeth of spur wheels by removing the interspaces. *a* is the relief angle or bottom rake, a side rake being provided by cutting the profile in an arc eccentric to that of the point path when rotating. Thus *b* is the centre for formation of the cutting tooth surface, while *c* is the centre of rotation. Now *d d* and *e e* are curves struck from *b*, and sections on each of these lines would be rectangular, but a section on *d e* must take the shape shown at *f*, because $d_1\ d_1$ is greater than $e_1\ e_1$ as seen in the end view. But as *d e* is the path of the point *d* during the revolution of the cutter, clearance or relief angle is therefore given at the side, and the cutter is said to be 'backed-off.' Of course this method can only be used with cutters of tapering profile; it enables us to preserve both form and width of cutting tool, however much is removed from the face, and is an improvement on the old cutter, which became



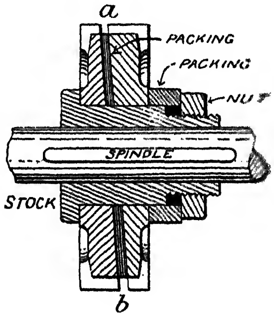
Spiral Mill
Fig. 181.



Grooving Mill Fig. 182



Twin Mills.
Fig. 184.



Expansible Mill.
Fig. 183.

narrower on re-grinding. The space between the teeth is to admit an emery wheel for grinding the faces.

Angle of tooth, although important, is still rather in dispute, principally because the same cutter, to avoid expense, is being used for various materials—a wrong procedure, without doubt. Probably some variation on the angles already given is necessary, because of the higher speed of cut. Experience seems to suggest the following :—

Cutting angle	80° to tangent.
Angle of relief..................	10° to tangent.
Front rake	10° to radius.

giving a tool angle of 70°. Small mills are made with radial teeth, corresponding to a cutting angle of 90°. A side rake of 10° should be given, and the teeth cut spirally or obliquely on a finishing tool.

Speeds.—There is still more variation in practice regarding these. They can be considerably higher than for other tools, because each tooth is in contact for only a small portion of the revolution, and has ample time to cool. The result is the more highly finished work that has brought milling into favour. The following speeds give the result of experience, and are fairly correct:—

Milling Speeds in feet per minute; and revolutions per minute, in terms of radius (r)″ of cutter.

	Roughing Cut.		Finishing Cut.	
	Ft. per M.	Rev. per M.	Ft. per M.	Rev. per M.
For Steel	30	$\frac{57}{r}$	40	$\frac{76}{r}$
,, Wrought Iron	40	$\frac{96}{r}$	55	$\frac{105}{r}$
,, Cast Iron	60	$\frac{114}{r}$	75	$\frac{143}{r}$
,, Gun Metal	80	$\frac{152}{r}$	100	$\frac{190}{r}$
,, Brass	100	$\frac{190}{r}$	120	$\frac{228}{r}$

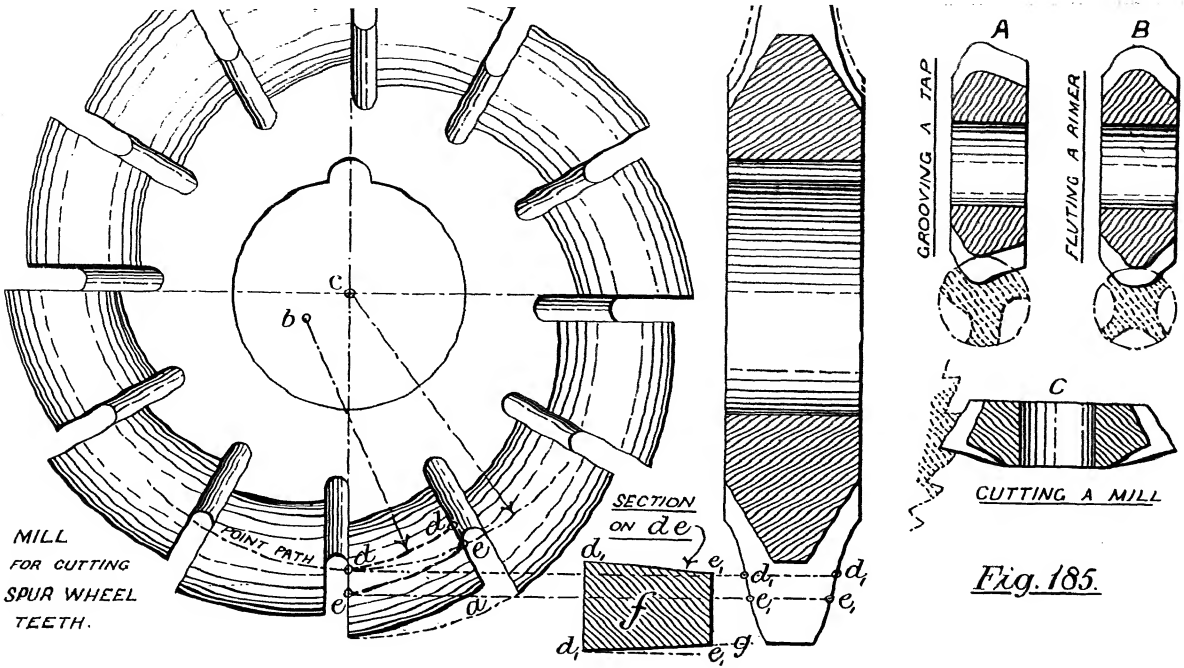

MILL
FOR CUTTING
SPUR WHEEL
TEETH.
POINT PATH
SECTION
ON de
GROOVING A TAP
FLUTING A RIMER
CUTTING A MILL
A
B
C

Fig. 185.

The *Universal Milling Machine* was of American design in the first place, and one of these useful machines is shewn in Plate XII., as made by Messrs. Tangye.

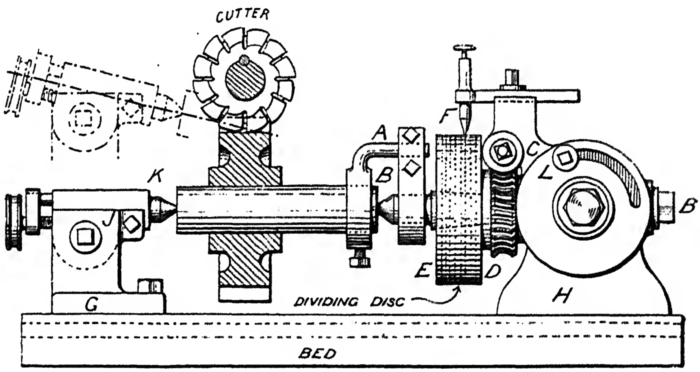
The mandrel A is driven from the cone pulley B, either directly or through the back gear, the latter being thrown out by the handle C, which turns eccentric bushes as usual. The mandrel is of large diameter, for stiffness, and revolves in coned bearings D D, the thrust when using a face cutter being taken by the steel tail pin E. A strong overhanging bracket F carries a small head H and centre G, to support an edge cutter, which centre is roughly adjusted by unbolting H, and finely by unscrewing the check nuts. The bracket is usually made round, and that form has some advantages, but is not so steady. The mill is either supported between centres, and driven from the catch plate; or has a shank similar to that described for the drill sockets at Fig. 169, when it is further steadied by the outer centre G; the latter is the more common method. A twin mill is shewn in position. Sometimes tools are fixed in the holes shewn in the catch plate at J, which is thus transformed into a face cutter, but the points must all be placed in the same vertical plane, so that each may take its proper share of work.

A vertical slide K, having square edges for rigidity under heavy cuts, supports a knee bracket L, which carries the table M, and between L and M are two slides N and P, the first for longitudinal, and the second for cross traversing. These swivel on the circular table Q, formed by their common surfaces, and P is made of extra length in plan to steady the table, a detail often neglected. A special point is the improved means of traversing the table. This is often effected by telescopic shafts with universal joints connected to the *end* of the table, and these sometimes act at such bad angles that the joints in crossing centres cause a slight dwell, which is reproduced on the work. This is avoided in the machine illustrated. A small cone pulley R on the mandrel drives the lower pulley S, keyed to the worm shaft T. This shaft carries a worm, gearing into a worm wheel *g*. A telescopic shaft U is connected to the inside of the worm wheel by a universal joint, and to the mitre wheels V W by a corresponding joint; these convey the motion to the screw X, which gives a cross feed to the

table. They are fixed in the centre of the swivelling table, and will transmit the feed motion with steadiness, even when the table is swivelled up to 45°, say for cutting spiral mills, twist drills, &c. By moving the hand lever Y the mitre wheel W may be drawn out of gear, and the cross feed given by hand, if desired, a catch Z ensuring the contact of the wheels when in gear. The longitudinal feed from screw *a* is rather a setting motion, there being few cases where other than a cross feed is desired.

The handle *b* is to raise or lower the table, which it does by turning the screw *e* through the medium of the worm gear *d*.

Other forms of machine are Vertical Milling Machines and Profiling Machines. In the former the cutter spindle is vertical, and a circular feed, as well as traverse, is given to the table. The latter is a smaller tool, where a vertical mill is traversed by a hand lever so as to accommodate itself to intricate forms. Good lubrication is necessary for all mills, and should be supplied under pressure from a small pump. (*See also pp.* 752, 811, 1020, *and* 1025.)



Dividing Headstocks.
Fig. 189.

Dividing Head.—When milling teeth of wheels, cutters, rimers, &c., the work is supported in centres shewn in Fig. 189, which are fastened to a small bed and bolted to the machine table. The wheel to be cut is fixed on a mandrel, and held in position

PLATE XII.

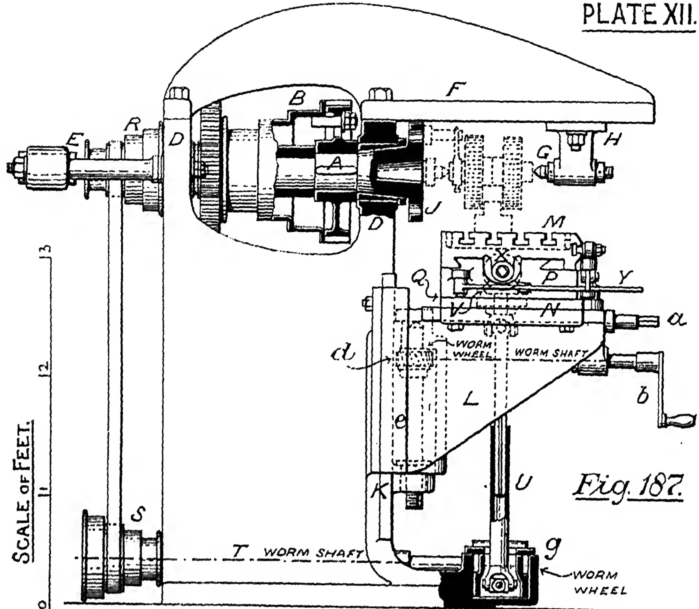


Fig. 187.

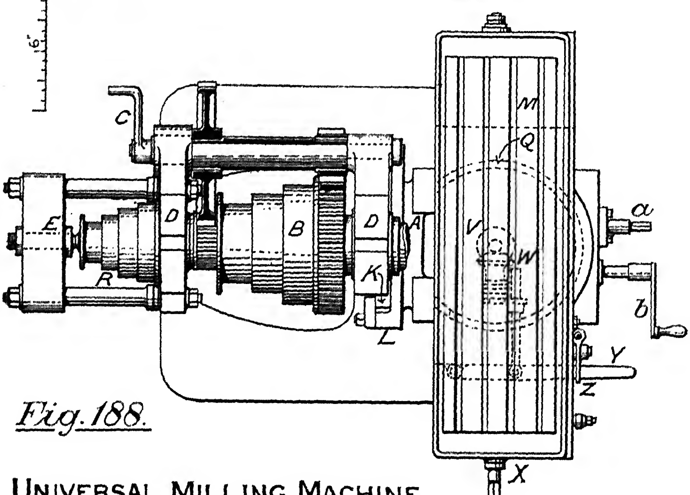


Fig. 188.

UNIVERSAL MILLING MACHINE.

(by Tangye's Machine Tool Co.)

face p. 180.

by the carrier A, which is screwed on the right hand centre, fixed in spindle B B. The spindle B B may be turned through any desired angle by the worm C and wheel D. E is a steel drum provided with small holes, representing various exact divisions of its circumference, and the point F can enter any of these, so as to set the spindle in the desired position. Knowing the number of spaces in the wheel to be cut, or flutes to the rimer, drum E, called a dividing plate, can be placed in each position in turn, and a cut taken. The heads G and H can be either bolted directly to the table, or packed to any convenient height, to accommodate a larger piece of work; or H may be bolted to the

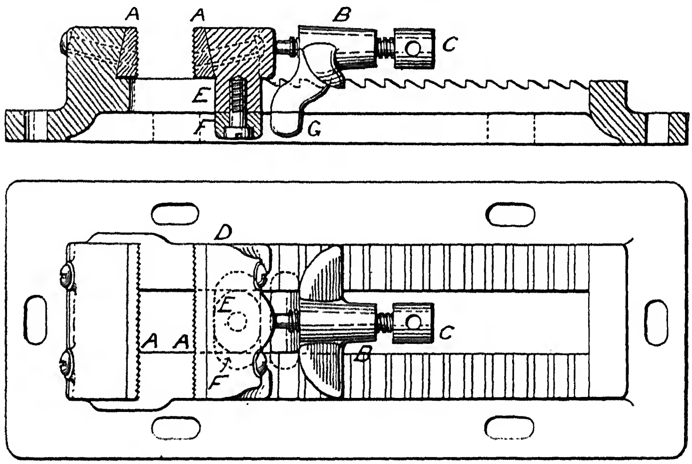


Fig. 190. Taylor's Machine Vice.

table, G packed, and the centres placed at an angle, as shewn by dotted lines, useful for tapered work. This is obtained by releasing the screws J and L, when centre K may dip, and B be tilted between the cheeks of H. B may be further turned at any angle up to the vertical, for milling cutters of various angles, and E has a conical socket to hold the mandrel supporting the work.

Similar centres are used when milling spiral cutters or twist

drills, but then the spindle must be rotated gradually, by change wheels connected with the feed.

The Machine Vice is a very useful appliance for Shaping, Milling, and Drilling machines. It is shewn at Fig. 190, and is bolted to the table of the machine, its object being the holding of work too small to be fastened down directly, or to facilitate the setting and re-setting of such work. A great desideratum is that the latter should bed firmly on the surface of the vice, accomplished in the example by the bevelled jaw plates A A, which pull the work down at the same time as it is gripped, by sliding on the bevelled surface. The nut B can be rapidly changed to any notch, and fine adjustment be given by applying a tommy to the screw C. The jaw D has a cylindrical shank and plate F; it can therefore be set at any horizontal angle, and the screw C will still bear upon it normally. B is also provided with lips at G, to resist upward pull.

CHAPTER VI.

MARKING-OFF, MACHINING, FITTING, AND ERECTING.

When an engine or machine is first projected, a rough 'general' drawing is made by the draughtsman, in order to determine the relation of the several parts; after which the 'detailing' takes place, which consists in drawing out each piece separately to a large scale, and at the same time classifying the work—putting all the forgings upon one set of sheets, and the castings upon others—so as to facilitate the distribution of the parts to the various shops and avoid delay.

Detail Drawings are fully provided with dimensions, and have red lines drawn round surfaces that need Fitting or Machining, viz., such as are required to fit or work together; and the Pattern Maker and Smith are thus enabled to decide where to leave extra material. It is the business of the *Marker-off* to 'line out' the rough work received from the above men; that is, indicate by a boundary line the amount of material to be removed by the *Fitter* or *Machinist*. The work is then finished and passed on to the *Erector*, who carefully puts it together to form the completed machine.

The Marker-off's Tools.—A large **plane table** or *Surface plate* is first required. This is shewn in Fig. 191, and its size varies with the average work to be lined-out upon it—from 4 ft. by 2 ft. up to 12 ft. by 4 ft. It is well ribbed underneath to prevent any possible distortion, and is planed very truly, being better also if filed up and a little scraping done upon it. The edges should be planed truly and adjacently at right angles, so that squares may be applied to them when necessary. Lastly, the feet should stand upon a firm bed of concrete, and be adjusted until the surface of the table is truly level, which often assists the marking-off considerably.

V blocks, to support cylindrical work upon the table, are

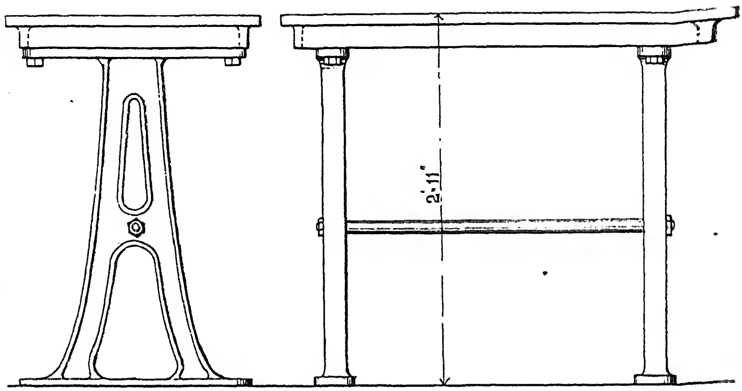
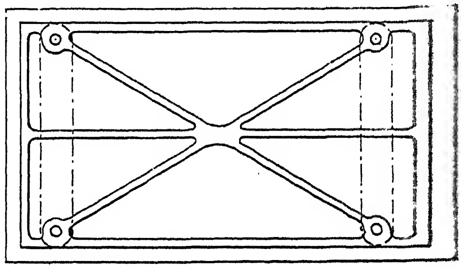


Fig. 191.
Marking-off Table.



underneath view

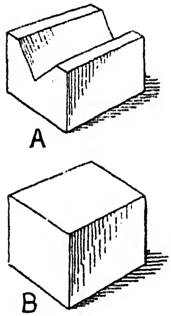


Fig. 192
Blocks.

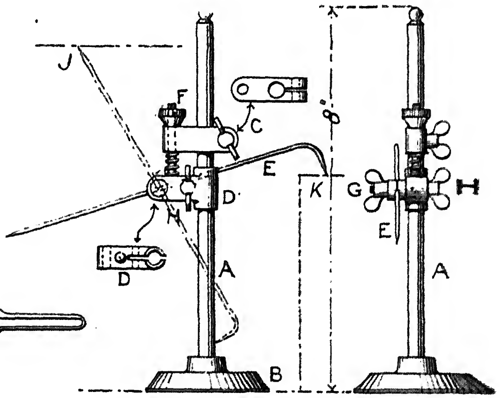


Fig. 193. Scribing Block.

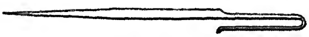


Fig. 194.
Scriber.

shewn at A, Fig. 192; and **Cubical blocks** are also provided, of several sizes, but each of known depth and so figured. They have their surfaces truly parallel, and are used to gain greater height for the Scribing block, as well as for the purpose of packing up the work (*see* B, Fig. 192).

The **Scribing Block**, Fig. 193, is a most important tool. It consists of an upright pillar A, fixed in a base B, which has been truly scraped underneath. Upon A slides the head D, which can be set to any height by tightening the nut H, a pointer or scriber E being at the same time fixed at any convenient angle by nut G. Most scribing blocks have no other adjustment, but in that shewn there is a screw at F for further accuracy; here the head C is first clamped, and D left free until finally adjusted by the screw F, after which D is firmly tightened and the scribing done. The scriber has one point straight and the other curved, the uses of these being shewn, where J can be made to 'scribe' a horizontal line on the work by moving the block along the table, and H may serve to 'feel' the height of certain other work. The scriber is of steel, well-hardened, and must be kept sharp by rubbing on an oilstone.

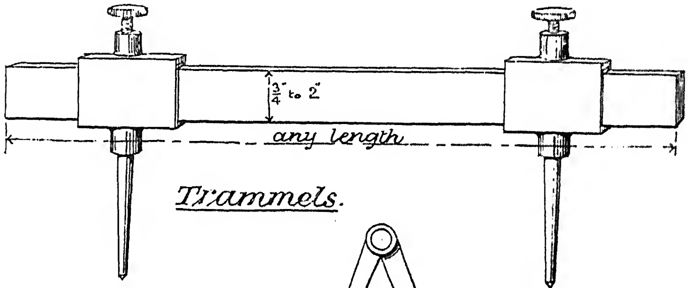
The **Hand Scriber** (Fig. 194) is to the marker-off what the pencil is to the draughtsman. It is pointed at one end, and hooked at the other for hanging to the pocket.

Compasses and **Trammels** must be provided for striking arcs of various radii, and as some pressure is required to make a sufficiently clear line on the work, both these tools should be sufficiently rigid; the former being supplied, for this purpose, with an arc and screw. Both tools are shewn at Fig. 195.

Accurate measuring **Rules,** with inches divided into eighths and tenths; **Squares** large and small (3 in. to 3 ft.); **Straight Edges** of different lengths; and **Callipers,** both for internal and external measurement, are all necessary tools; while if the work is too large to mark-off on a table it should be levelled, and all lines be drawn by reference to an ideal horizontal or vertical plane, necessitating the use of either a *Spirit Level* or the **Square and Plumb-Bob** shewn at Fig. 196, the latter being the only tool in favour with the best workmen, as levels are known to get out of order so easily.

Of **Centre Punches** two are required, the larger for marking main centres only, and the smaller, or *Dotting Punch*, for the purpose of making a scribed line more lasting and apparent, by marking a series of punches or 'dots' along its length.

A light crane arm and Weston block is also of use when work of large size is to be manipulated.



Trammels.

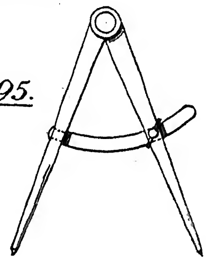


Fig. 195.

Compasses.

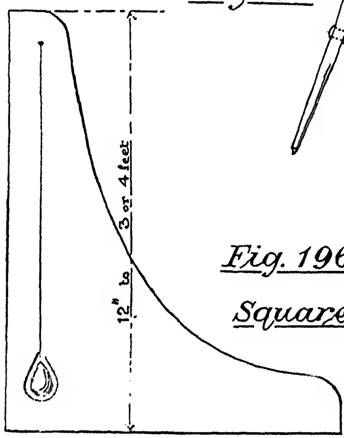


Fig. 196.

Square with Plumb-Bob.

√ **Fitter's Tools.**—Most fitting is done at the bench, the work being gripped in a vice, of which there are two principal kinds, 'Leg' and 'Parallel.' The old-fashioned **Leg Vice,** made of wrought iron with steel-faced jaws, is still considerably used, because capable of withstanding a large amount of hard

[illegible] It is shown at Fig. [illegible], [illegible], which the [illegible] long [illegible] parallel [illegible] the vice [illegible] the **Parallel Vice** [illegible]. It may have either [illegible], [illegible] *instantaneous grip*. [illegible] latter class, and instead of [illegible] forming a toggle joint. Referring to the [illegible] the toggle bars, a, being pivoted against the [illegible] b, which is capable of [illegible] the teeth in the sliding bar c. The bars are further held together [illegible] by the spring d. In order to grip a piece of work, the handle is first thrown back as in the figure, and the bar b, being free, is pushed *nearly* up to the work. The handle is then pulled towards the operator. When it reaches the position e, the bar a is released, and the spring d brings the teeth of a and b into contact. Then, as the handle is pulled further forward, the eccentric boss f acts on the back of the lever a, and, nearly straightening the toggle, urges the bar b forward with great force.

The proper height to place the vice is an important consideration, and depends on the class of work for which it is to be used. If this be light, the jaws should be rather higher than the elbow, to bring the work nearer the eye; but if the work be heavy, the fitter needs to put his whole weight on the file, and the jaws are then placed rather lower than the elbow. A good average is [illegible] ins. to the top of the jaws, which requires a bench [illegible] ft. [illegible] ins. high. The **Hand Hammer** has a head of about [illegible] to [illegible] lbs. weight (the latter only for very heavy chipping), and a shaft from [illegible] to [illegible] ins. long. It is shown at Fig. [illegible]. The 'face' is flat, but the 'pane' is usually spherical, for riveting purposes.

The fitter's chisel, called 'cold,' or 'chipping,' has three varieties. The **Cross-cut Chisel**, Fig. [illegible], is for roughing out work for the flat chisel to follow upon. When in use it [illegible]

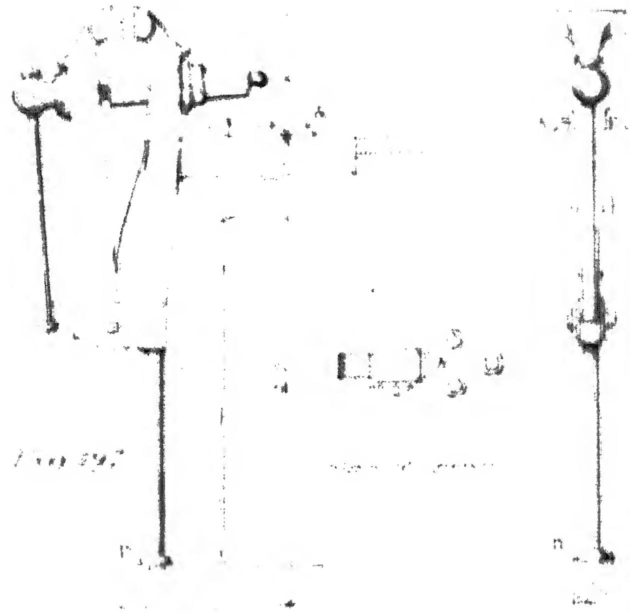


Fig. 297.

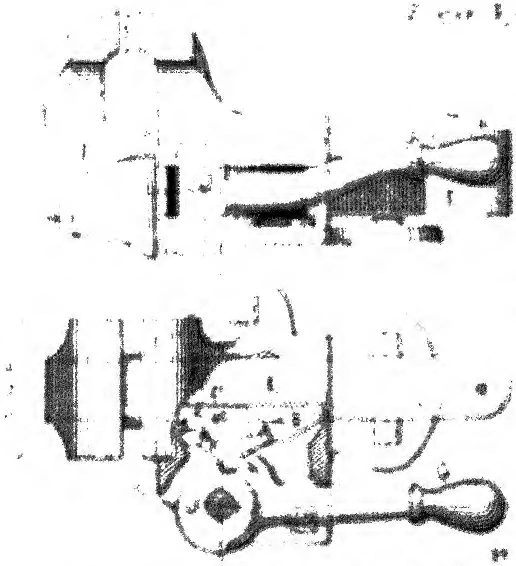


Fig. 298.

Stephens' Parallel Vice.

the elevation, and B the plan view. The **Flat Chisel**, Fig. 201, is used to true up surfaces previous to filing; and the **Round-nosed Chisel**, Fig. 202, is for chipping out concave flutings; but the last is more of a machinist's than a fitter's tool, the lathe-man and driller both using it for 'drawing' a centre-punch mark or countersink, which has been begun untruly, by chipping a little to one side of the depression so as to alter the position of the centre, after which the drill or square-centre is again applied.

The point of a flat chisel is ground symmetrically on each side, and should enclose an angle about equal to that of the **V** screw-thread, viz., 55°, though a slightly smaller angle may be used in finishing. After chipping, the surface must be further trued by filing.

Files may be classified in two ways: (1) by the *contour*, both in length and in section; (2) by the kind of cut and degree of fineness. The length must also be stated, measured along the edge, not including the tang. The cut may be double or single, the latter being also called 'float' cut, but as this is principally used for saw files, it will not be considered further. Longitudinally, files may be *parallel* or blunt, and *taper* or pointed; and in cross section they may be *flat*, *three square* or triangular, *half-round*, *round*, and *square*. The fineness of cut is represented by the terms *rough*, *middle*, *bastard*, *second-cut*, smooth, and *dead-smooth*, the last four only being required by the Fitter. *Safe-edge* files are those left uncut on one narrow edge, to serve in filing a surface near a corner, without destroying the truth of that at right angles to it. Files are either machine or hand cut, of which the latter are most in favour. It will be seen therefore, from the previous information, that a particular file may be described something as follows:—'12 in. hand, taper, flat, bastard, double-cut, safe-edge file.' As the teeth only cut in one direction the file is analogous to a planing tool.

Scrapers still further true up a surface left by the file or machine tool. They are made from old files, by grinding off the teeth and sharpening the edges, and have three principal shapes as shewn in Fig. 203: *Half-round* (A), useful in scraping a bearing; *three-square* (B), sharpened on the long edge for truing up

Fig. 199.
Hand Hammer.

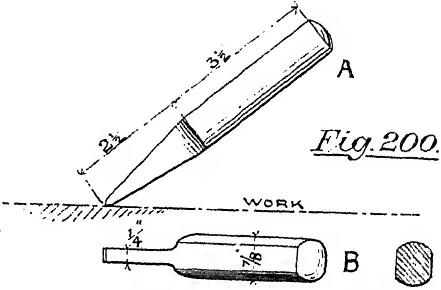


Fig. 200.
Cross-cut Chisel.

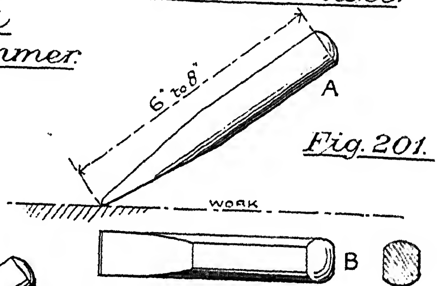


Fig. 201.
Flat Chisel.

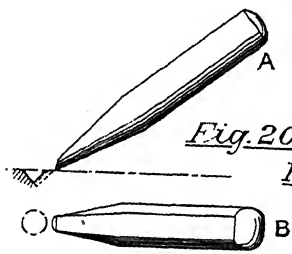


Fig. 202.
Round-nosed Chisel

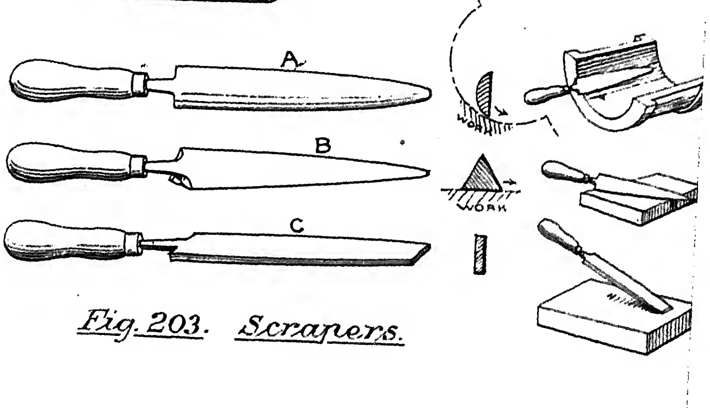


Fig. 203. Scrapers.

[illegible] and [illegible] on the end [illegible]

[illegible] must be referred [illegible] should possess two [illegible] one (A, Fig. 204), of

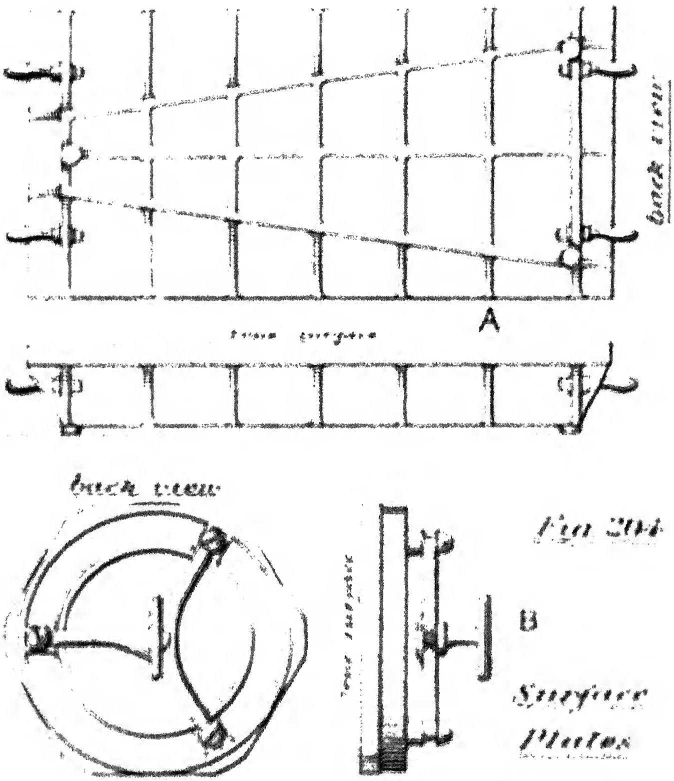


Fig. 204
Surface Plates

moderate size, so as to suit the average work to be *laid* upon it, and a *hand* plate (B, Fig. 204), to be moved *over* the work, which, being supplied with a removable handle, may serve as a hand plate when needed. Their method of use will be treated later (*see* p. 210).

Lastly Screwing Tackle is required, which consists of **Stocks**

and Dies for bolts or spindles, and taps for the nuts in which these spindles are to fit. **Taps** are made in sets of three to each diameter of screw, and to cut **V** threads of 'Whitworth' pitch; that is, whose pitch per diameter agrees with those in the table devised by Sir Joseph Whitworth, and which is here appended:—

V Threaded Screws (Whitworth).

Dia. in ins.	Threads per inch.	Dia. in ins.	Threads per inch.	Dia. in ins.	Threads per inch.	Dia. in ins.	Threads per inch.
$\frac{3}{16}$	24	1	8	$2\frac{1}{4}$	4	$4\frac{1}{2}$	$2\frac{7}{8}$
$\frac{1}{4}$	20	$1\frac{1}{8}$	7	$2\frac{1}{2}$	4	$4\frac{3}{4}$	$2\frac{3}{4}$
$\frac{5}{16}$	18	$1\frac{1}{4}$	7	$2\frac{3}{4}$	$3\frac{1}{2}$	5	$2\frac{3}{4}$
$\frac{3}{8}$	16	$1\frac{3}{8}$	6	3	$3\frac{1}{2}$	$5\frac{1}{4}$	$2\frac{5}{8}$
$\frac{7}{16}$	14	$1\frac{1}{2}$	6	$3\frac{1}{4}$	$3\frac{1}{4}$	$5\frac{1}{2}$	$2\frac{5}{8}$
$\frac{1}{2}$	12	$1\frac{5}{8}$	5	$3\frac{1}{2}$	$3\frac{1}{4}$	$5\frac{3}{4}$	$2\frac{1}{2}$
$\frac{5}{8}$	11	$1\frac{3}{4}$	5	$3\frac{3}{4}$	3	6	$2\frac{1}{2}$
$\frac{3}{4}$	10	$1\frac{7}{8}$	$4\frac{1}{2}$	4	3		
$\frac{7}{8}$	9	2	$4\frac{1}{2}$	$4\frac{1}{4}$	$2\frac{7}{8}$		

Very rarely are taps used beyond $1\frac{1}{2}$ ins. diameter, larger sizes being screwed in the lathe. The set of three is shewn at Fig. 205, and includes 'taper' (A), 'middle' (B), and 'plug' taps (C).

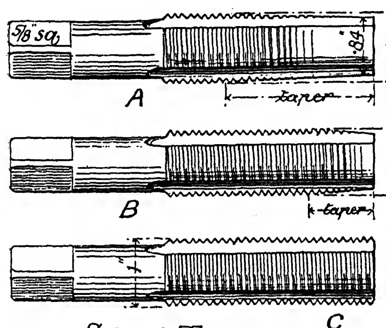
These are made by forming in the lathe a perfect screw thread upon a 'blank,' and afterwards fluting to the section shewn enlarged at E, so that when the tap is turned right-handed it has a cutting angle of 90°, and a small relief or clearance angle, removed with the file. Next, two-thirds of the length of the taper tap, and one-third of the middle tap are turned off, after which all are hardened as shewn at page 127. When in use the nut must first be tapped, and the bolt afterwards screwed to fit it. After drilling to 'tapping size,' that is, to the diameter at the bottom of the screw thread, the taper tap is first entered (while the nut is held in the vice), and is turned round by a wrench D applied to the square on the top. Only when turned right-handed is the thread cut, as will be seen on reference to E,

Fig. 205 ; and a left-handed turn will release the tool. When the taper tap has done its work the middle tap is introduced in like manner, carrying the operation a little further, and finally the plug tap is passed through to give the finishing cut. After every stroke forward, the workman releases the tool slightly, so as to avoid undue pressure and perhaps breakage. (*See p.* 1025.)

A stock and dies is shewn at **Fig. 206.** A is the stock, provided with handles for turning, and B is an enlarged view of one of the dies, having a thread upon it in reverse, and four cutting surfaces at 90°, two to each direction of rotation: so that the thread may be cut both on advance and return. The dies are shewn in position in the stock A, being dropped in at *e* and slid along : then tightened by a tommy applied to the screw *d*. The bolt to be screwed is first turned to the outside diameter of the tap, and then fixed in the vice. The dies are separated slightly, the stock brought over the bolt as at C, and the screw advanced. The stock is now rotated until the length of the bolt is traversed; then, on reversing the motion, a slightly increased pressure given to the dies ; and so the bolt is re-traversed again and again, until so cut into by the dies as to show a perfect thread, and gauge to proper diameter, which may be proved by trying upon it the already tapped nut, and any degree of tightness obtained after such trial. At each stroke a slight backward release is given as before, and oil may be used as a lubricant. Various sized dies may also be applied to the same stock.

For screws under a $\frac{1}{4}$ in. diameter the **Screw Plate** in Fig. 207 replaces the stock and dies, and only one tap is required instead of three.

The pitch of a screw being measured lengthwise from centre to centre of the threads, let us unwind the latter, both at the top and bottom of the **V** groove. The diagram in Fig. 208 will shew the result obtained in each case, and it will be clearly seen that the angle at the bottom of the thread is larger than that at its top. But the action of the dies, in cutting, is to first mark out the top of the thread with that part of the die formed to finish the angle at the bottom, and it follows that by the time the thread is finished, there will be an unnecessary endlong play of the bolt in the nut. These faults are somewhat avoided by the use of the



Set of Taps.

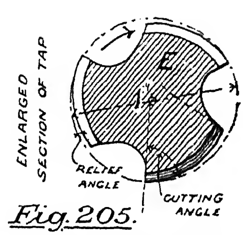
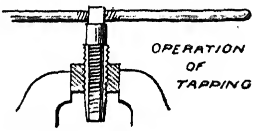
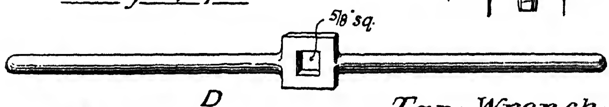


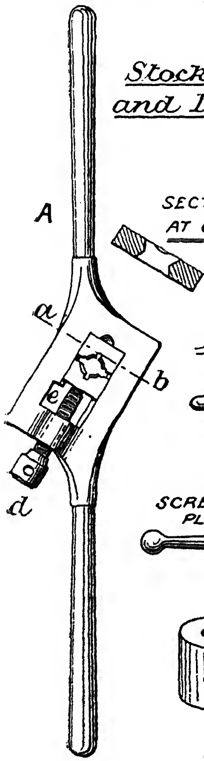
Fig. 205.





Tap Wrench

Stock and Dies.



SECTION AT ab.

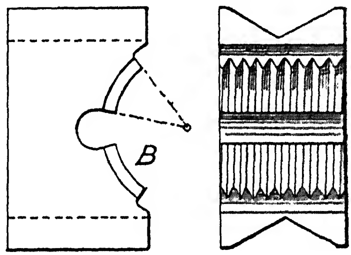
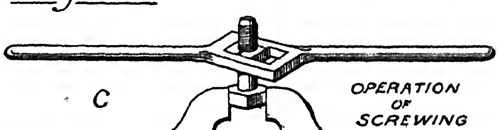


Fig. 206. ENLARGED VIEW OF DIE.



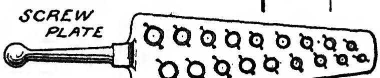


Fig. 207.

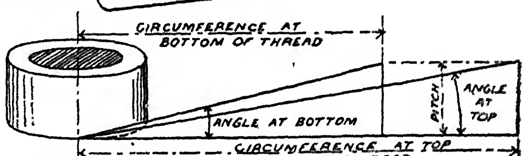


Fig. 208

Whitworth **Guide Screwing Stock** in Fig. 209. Here there are three dies, *a* being the 'guide,' cut so that its ridges just fit the bolt *at first*, and are made to mark out the correct angle for the *top* of the thread. *b* and *c* are the cutting dies (gradually advanced by the wedge bolt *d*), and these ultimately give the correct form for the *bottom* of the thread. But the only perfectly true method of cutting a screw is by means of the lathe, where the tool is fixed in the slide rest and the thread formed by the gradual advance of the rest coupled with the rotative movement of the work. (*See Appendix II., p.* 815.)

Machinists' requirements, in addition to the tools mentioned in Chapter V. These consist principally of grinding and sharpening tools.

The **Grindstone**, though banished from some shops in favour of emery, is still so extensively used as to deserve mention. It is shewn at Fig. 210, and the stone fits on a square spindle having journals at the ends, lying in simple bearings. Large washers are placed on each face of the stone and the nut *a* tightens these. Fast and loose pullies are provided for driving by power, and a shield *b* to prevent the water flying about, the latter being a necessary lubricant. *c* is a rest for the work, placed rather high up, and as close to the stone as possible, to avoid accidents. The direction of rotation of the stone is shewn by the arrow, and the speed is such as to give from 800 to 1000 ft. per minute surface velocity. It is not advisable to actually run the machine in water as this tends to soften the stone.

The **Emery Grinder** is seen at Fig. 211. Its bearings are longer than those of the grindstone, and its peripheral velocity much higher, being about 5000 ft. per minute. A plentiful supply of water is required for tool sharpening, otherwise, with most emery wheels, the temper would be drawn and the wheel become glazed. The water is shewn in the figure as coming from a vessel above the wheel, but is sometimes supplied under pressure from a small pump. Glazing is caused by the cementing material becoming softened by the heat produced in grinding, though properly the cement should wear gradually and fall away with the emery powder. A very useful form of emery grinder is shewn in Fig. 212, suitable both for tool grinding on the larger

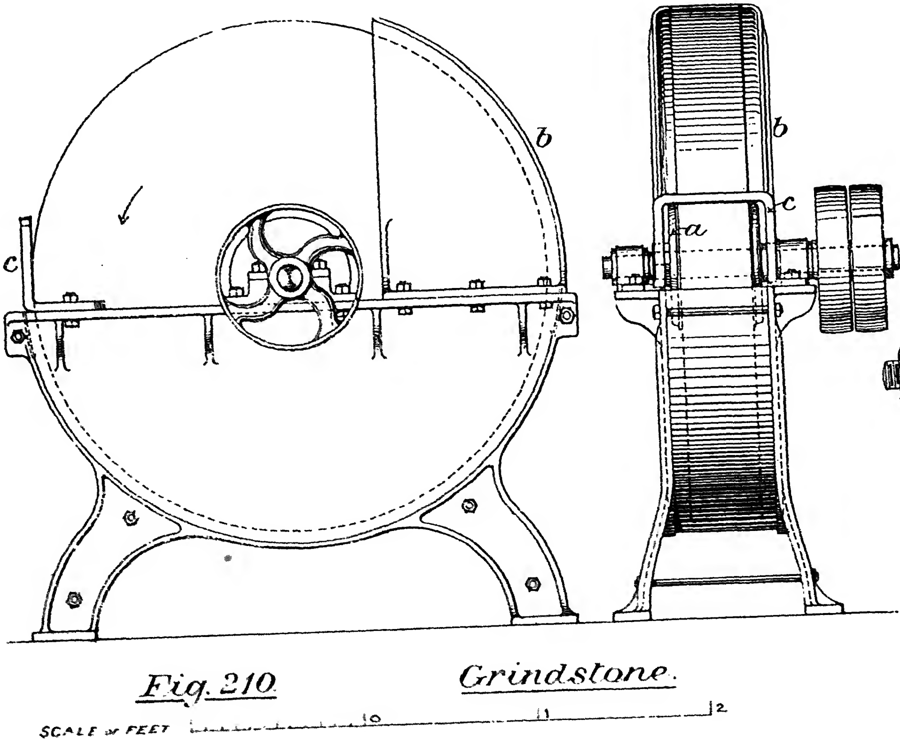
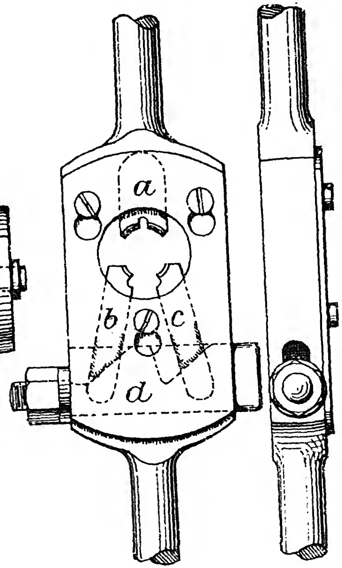


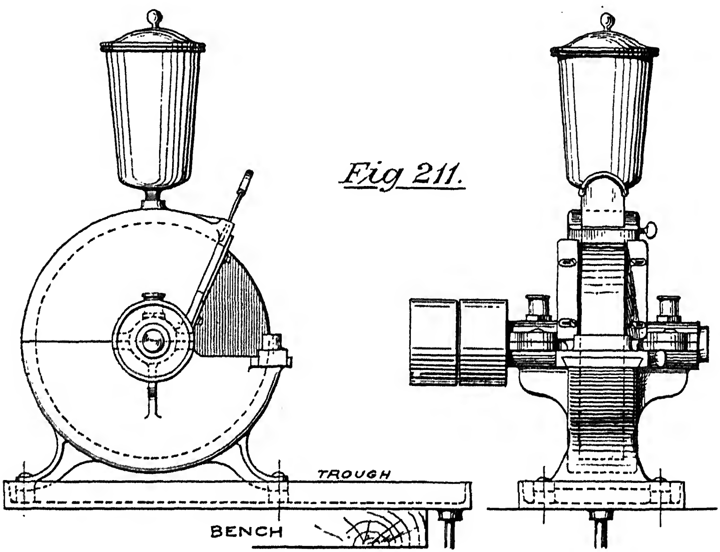
Fig. 210. Grindstone.



Whitworth's Guide Screwing Stock.

Fig. 209.

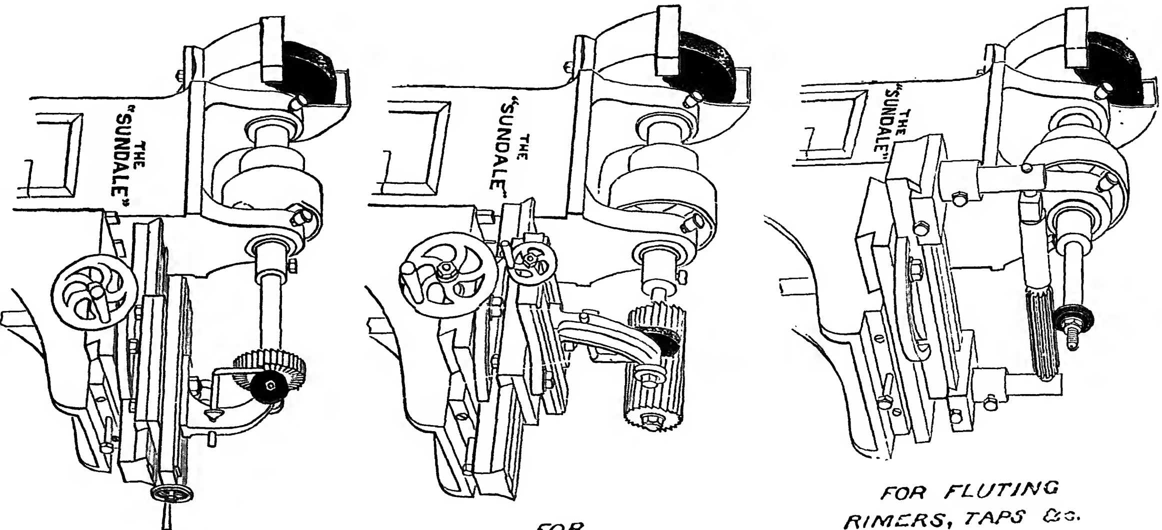
wheel, and fluting, &c., on the small wheel. It is made by Messrs. Selig, Sonnenthal & Co., and a small attachment is provided to carry the wheel when grinding milling cutters, the latter being then held on the spindle of the machine, and the



Emery Tool-Grinder.

wheel driven by catgut band. Emery wheels may be used for general grinding and removing of surplus material and thereby save a large amount of fitting. (*See p.* 1030.)

Twist-Drill Grinder.—These are of various designs, the one in Fig. 213 being made by Messrs. Selig, Sonnenthal & Co. The end of a twist-drill would be conical in shape but for the clearance or relief angle. The true surface becomes, accounting for clearance, a cone having a helix for its base, and enclosing an angle of 118°. A section of this cone, then, made at right angles to one of the slant sides, would give a curve deviating slightly from a hyperbola, due to the clearance. We will now examine the method by which this hyperbolic surface is ground in Messrs. Selig's machine.



FOR CUTTER SIDES

FOR CUTTER EDGES

FOR FLUTING RIMERS, TAPS &c.

Universal Milling-Cutter Grinding Machine.

BY SELIG, SONNENTHAL & Co.

Fig. 212.

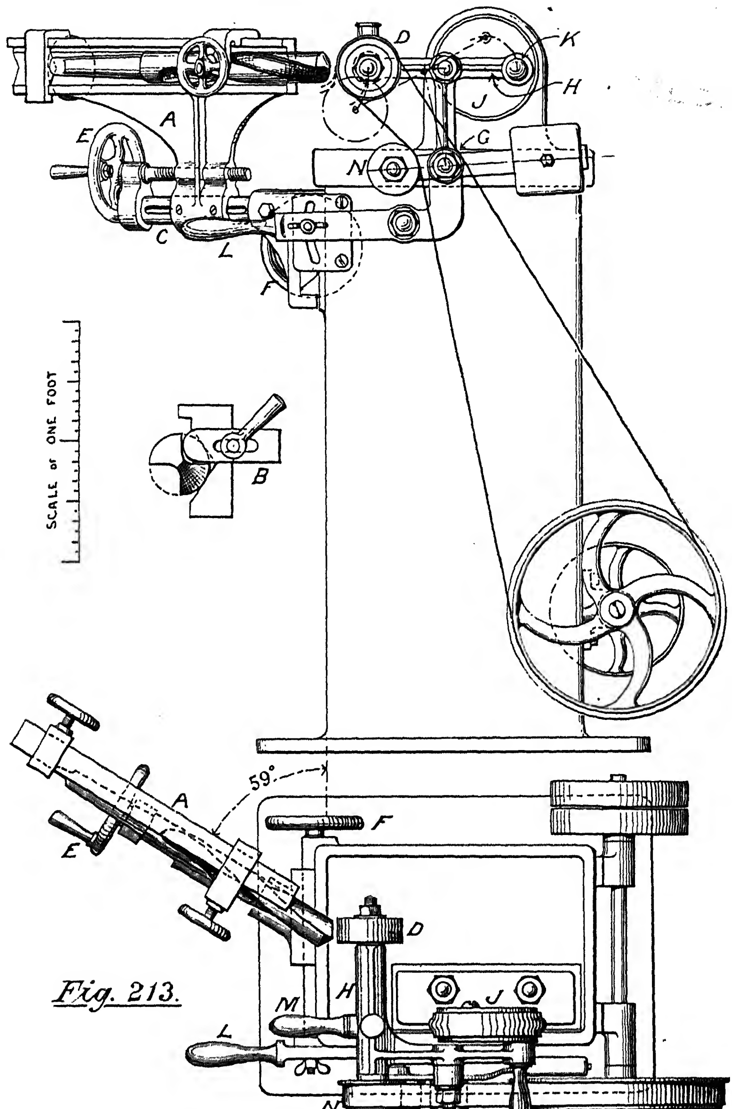


Fig. 213.

Twist Drill Grinder

(By Selig, Sonnenthal & Co.)

First the drill is clamped in a **V** groove made in the support A, and is held in the proper position by means of the plate B placed at the front end of the groove. The support A rides on a guide-arm C, which, in plan, is set at an angle of 59°, or half the angle of the drill. This allows the surface of the drill point to lie parallel to that of the emery wheel D. The hand-wheel E serves to bring the drill to the wheel, and F turns a screw for the purpose of taking up various surfaces of the wheel so as to produce equal wear. G is a fulcrum, supporting a rocking arm, which, in turn, carries a horizontal arm H. One end of H encloses the emery wheel spindle, and the other is pinned to the rotating disc J. It follows, therefore, that if the disc J be turned left-handed by taking hold of the handle K, the rocking arm will deviate to the front, and the centre of the emery wheel will describe the approximate hyperbola required to be ground off the drill point, as shewn by the dotted lines in elevation. By fixing the fulcrum G at slightly varying heights by means of the hand lever L, it is possible to obtain sufficient variation in the drill curve to suit various sizes of drills; and, as the driving strap is changed in position, it is kept tight by the jockey pulley N provided with a balance weight. When using the machine the workman takes hold of the handles M and K, and pulls K towards him, and after one surface of the drill has been ground the latter is turned round in the **V** groove, and the opposite surface trued up, B then serving to register the second position with the first.

The Capstan or Turret Head.—Although we were supposed to have completed our descriptions of machine tools in Chapter V., our work would be incomplete without an account of this very important labour-saving appliance. The lathe in Fig. 214 is shewn supplied with both Capstan-Head Slide Rest, and Screw-Copying apparatus, and is designed by Selig, Sonnenthal & Co. A is the head, which is capable of holding six tools, to be used in succession on the work in the lathe. These are placed in position by releasing a catch E, turning the head by hand, allowing catch E to return to its place by means of a spring, and finally clamping the rest firmly by means of the lever D; all this occupying but a very short space of time. Of course, it may often be necessary to use both slides to put the tools in position, as will be seen, and the

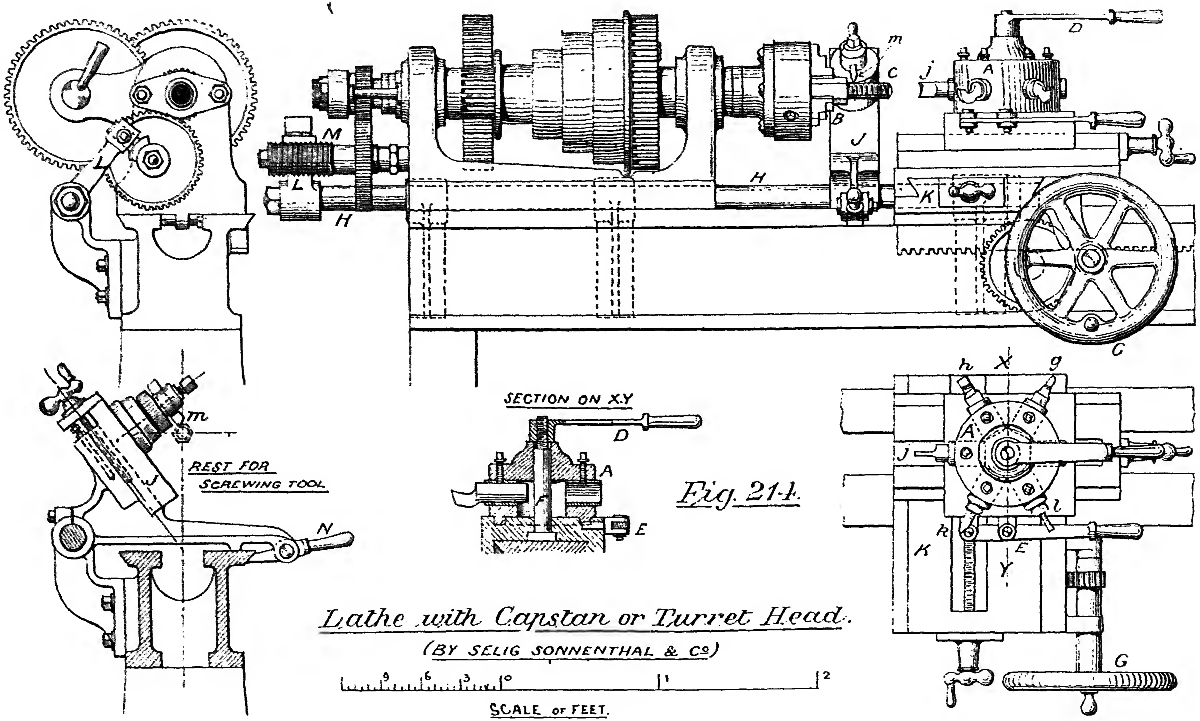


Fig. 214.

Lathe with Capstan or Turret Head.

(BY SELIG SONNENTHAL & Co)

usual rack is provided for moving the saddle scme distance along the bed.

Turning now to the screwing gear, J is a rest for the screwing head, with screw for adjustment, and when in position for work is held by the handle N. At the same time a lever L, provided with a screwed die fitting in the threads of the screw M, is placed at the other end of the shaft H, so that when the screwing tool is on the work, L engages with the screw M, but if the rest J be lifted and thrown back, L is at the same time released. When in operation the screw M is rotated from the mandrel by gearing of 2 : 1, so that a screw is cut at C, having half the pitch of the copy and of reverse hand, M being usually left-handed, and C right-handed. Of course the shaft H is capable of longitudinal motion, and the piece M, being hollow, can be removed, and another of different pitch applied, while the die, usually made of copper or soft brass, does not need special cutting, but will find its way into the threads of the screw.

Lastly, the lathe is provided with a hollow mandrel, which is very useful for small articles that can be cut from a continuous bar. An example of such work is shewn in progress, being the making of a small tap bolt. A hexagonal bar is held in a concentric chuck, drawn forward to a convenient length, and the *roughing* tool *g* first applied, traversing to the front for position. The bolt being roughed down, is *finished* by the tool *h*, and has its end *rounded* by *j*. Next the *screwing* is performed by bringing over the tool *m*; and, lastly, the *chamfering* and *parting* are done respectively by the tools *k* and *l*. It will be, therefore, clear that a great deal of time and labour may be saved by the use of such a tool where articles have to be made in quantity. All bolts and studs are turned at such a lathe. (*See Appendices, pp.* 814, 978, 1040.)

✓ **Erector's Tools.**—These must include Lifting Tackle and a Portable Drilling tool. The latter is known as the **Ratchet Brace**, and is shewn at Fig. 215 in position for drilling a hole. The pillar A is clamped to the work, and carries an arm F, which can be set at various heights, to take the brace and drill B. As the latter is ground to cut in one direction only (see *d*, page 168), the brace is made to enclose a ratchet wheel C fixed to the drill

[illegible], which wheel is driven by the spring pawl D, on account [illegible] action when the handle is pulled towards the operator. [illegible]. The feed is given by holding [illegible] with the left hand, while the drill is turned right handed, [illegible] and screwed as before.

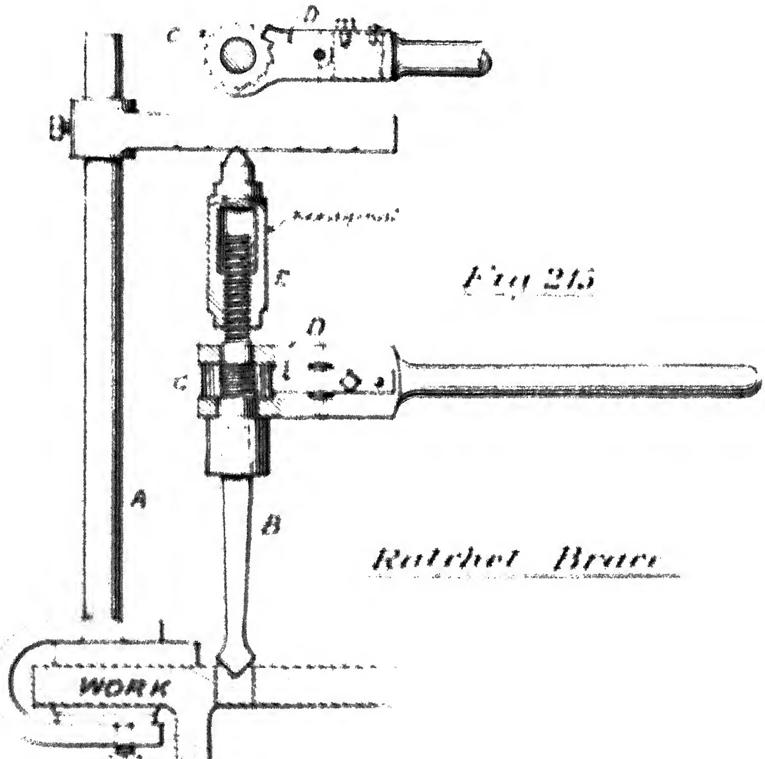
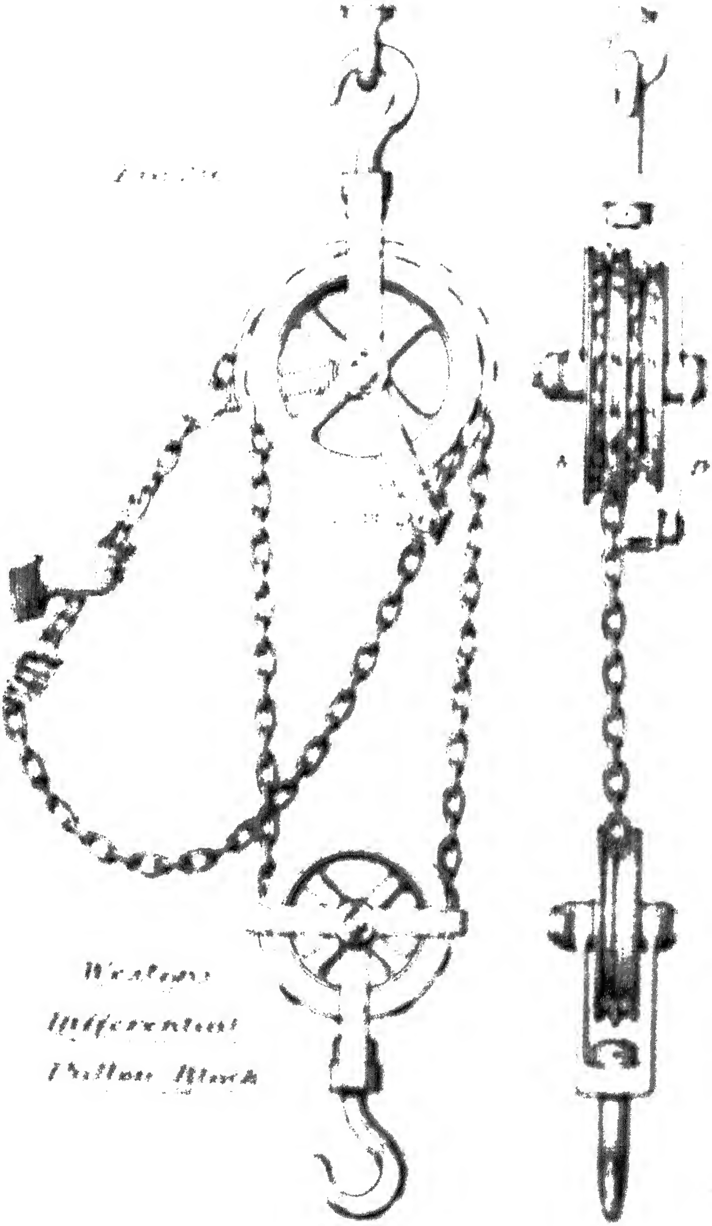


Fig 215

Ratchet Brace

As a simple [illegible] lifting gear, the **Weston Pulley Block** has stood its ground well. The principle is differential. The upper pulleys A B are cast together, and are slightly different in diameter. They are gripped by the chain, which lies in a specially formed groove, and while the upper pulleys are once rotated, the lower or movable pulley is raised by half the difference of the circumferences of A and B, thus giving a great



Weston Differential Pulley Block

mechanical advantage. There is a very large loss by friction (some 70%), but this resistance is useful as serving to sustain the weight when the chain is released by the hand. (*See p.* 1047.)

Jacks are useful where overhead support cannot easily be obtained. Fig 217 shews a simple **Bottle Jack,** the 'bottle' serving as a fixed nut in which the screw rises when turned by a tommy bar; and Fig. 219 represents a more powerful **Jack with worm gear.** Here the screw is prevented from rotating by the jaw *d*, and is, therefore, raised by the rotation of the worm wheel A, which acts as a nut. In the example a handle of 14 ins. radius turns, by means of a worm, a worm wheel of 16 teeth, enclosing a screw of 1¼ in. pitch; and a weight of 10 tons is thereby lifted. The lower jaw *d* is for loads that are near the ground, and the jack may be traversed, when in position, by the ratchet arm C, applied to the screw *b* at either end.

The **Hydraulic Jack** is both very useful and very interesting, and is shewn at Fig. 218. It has an upper and a lower jaw to suit various work, and both are part of the cylinder A. B is a reservoir in which is placed oil, or water and glycerine. The handle being moved upward on the fulcrum C, the pump plunger D is thereby raised, and the liquid enters the pump through the suction valve E; on the down stroke it is forced through the delivery valve F, and exerts a pressure behind the ram G, thus lifting the cylinder A. The valves are 'non-return,' being loaded by springs, and the ram is packed by a cup leather. It being required to lower, the screw-down valve H is released, and the liquid runs back to the reservoir. Screw J is for filling the latter, and K is an air hole to assist the pump suction. The power obtained depends both on the leverage and on the ratio of the areas of plunger and ram, and may be calculated in the same way as for the hydraulic press, which is discussed at *p.* 736.

There are a few other small tools of use to the Erector. The **D Cramp** A, Fig. 219*b*, is for temporarily fastening two pieces of work together; and the **Key Drift** B for releasing keys when fitting wheels upon shafts. The file C is provided with a special handle, usually made from a bent bolt, to enable a very large surface to be filed; and the **Square Drift** at Fig. 219*c* is really a Fitter's tool, being used to clean out square holes too small to be

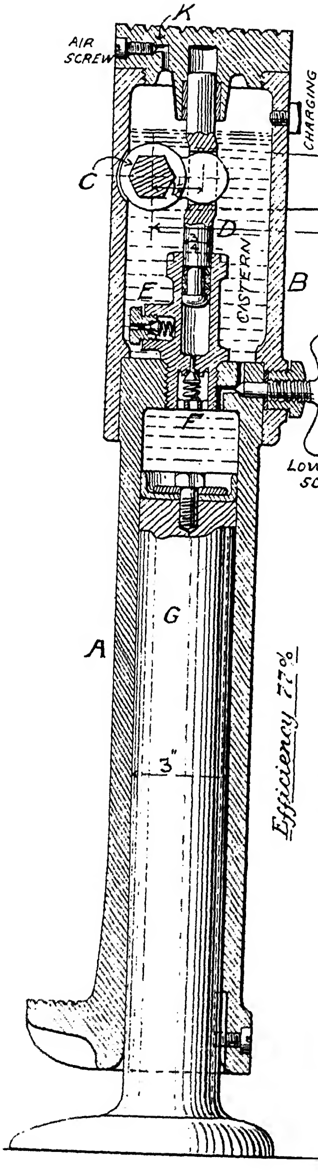
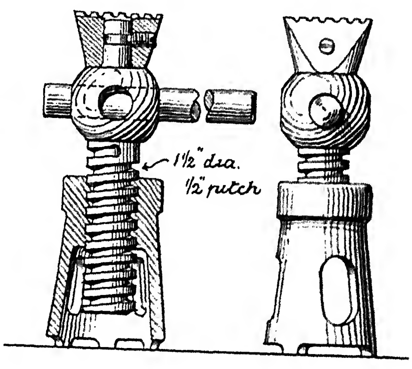


Fig. 218

20 Tons Hydraulic Jack

Fig. 217.



2 Tons Screw Jack

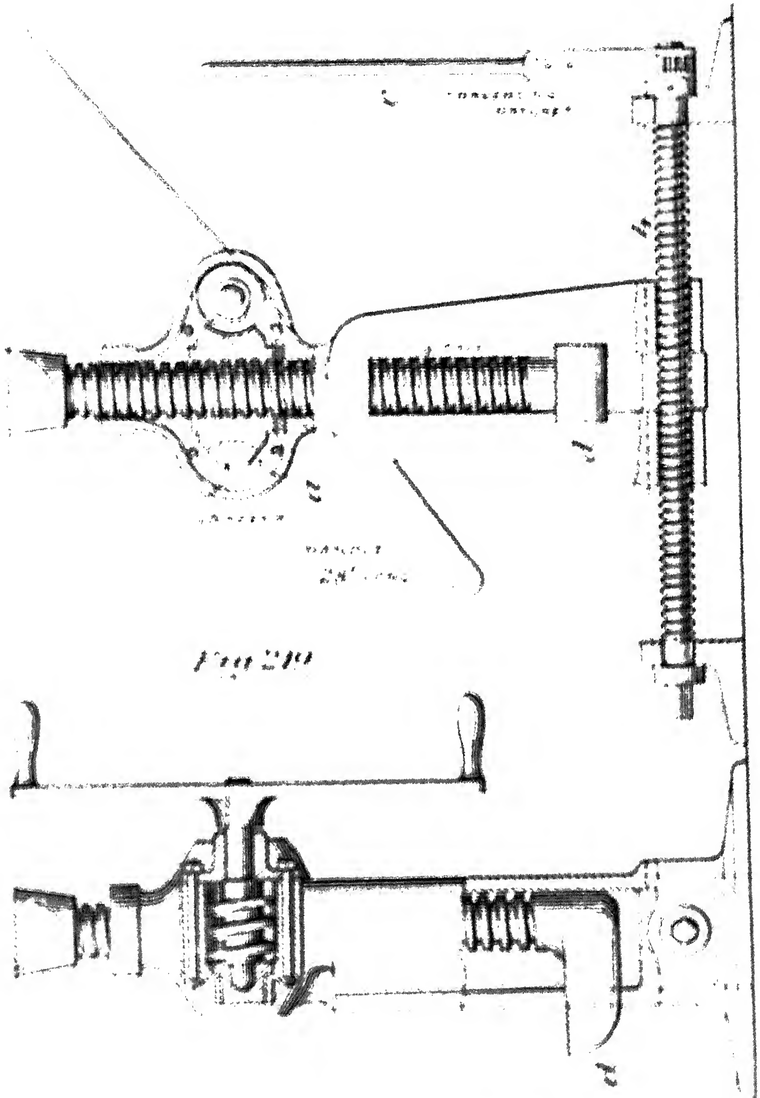
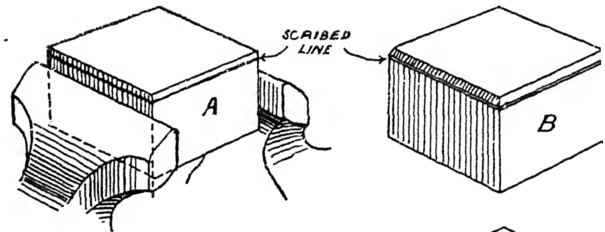


Fig 210

10 Tons Lifting Jack



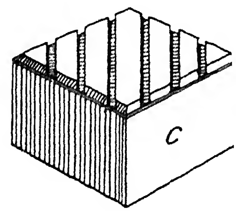
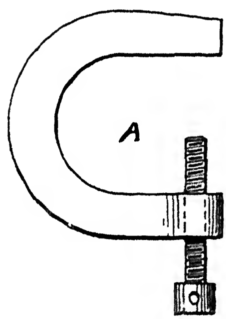
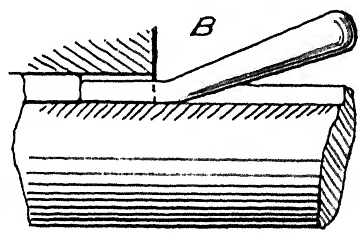


Fig. 219a

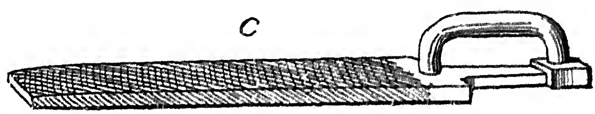
Chipping a Surface.



D CRAMP



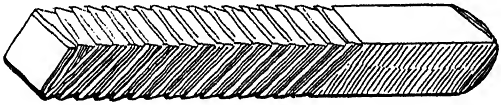
KEY DRIFT



FILE FOR LARGE SURFACES

Fig. 219. b.

drilled and slotted. A **Lead Hammer**, for use on finished work; a **Hack Saw**; and an **adjustable spanner** are also advisable. Round holes are cleaned by the **Parallel Rimer** in Fig. 231, and taper holes by means of a **Taper Rimer** similarly constructed.



Square Drift. Fig. 219c.

General Processes.—Chipping.—Although hand processes cannot well be taught on paper, a general idea may yet be obtained. We will consider ourselves provided with a cubical block of metal, and that it is desired to remove a rather large amount of material from one of the surfaces. We commence by placing the block on the marking-off table, and, chalking the edges, scribe a line round as shewn at Fig. 219*a*, to indicate the layer to be removed. This done we place the work in the vice and chip with flat chisel a chamfer along the edge of the block, *nearly* down to the scribed line, as at B, and make this fairly straight with a rough file. Now the cross-cut chisel is applied, and with it the cross grooves are cut as at C, each groove being tried with a straight edge, to make sure it is not carried too far below the general surface. We are now in a position to commence the removal of the strips that remain by means of the flat chisel, constantly trying the work with the straight edge, until the whole is as perfect as the chisel can make it. The position of the workman and the angle of chisel are shewn in Fig. 220, and practice only will shew the steepness of angle required for the deep cut, and the shallower angle for the lighter cut.

Filing.—The file is next applied, and the various 'cuts' used in order from bastard to smooth. True filing requires considerable skill, the tendency to the production of a convex surface being very great. The back stroke needs no pressure, as the teeth do not then cut; but during the forward stroke all possible

pressure is put on with both hands, and the file carefully guided in a perfectly horizontal direction, the position of the hands being shewn in Fig. 221. Comparatively narrow surfaces that are not to be scraped are generally smoothed by 'draw-filing,' the file teeth being rubbed with chalk to compel the small particles to drop out, and thus avoid the scratching of the work, and a still further polish given by means of fine emery cloth wrapped round the file. The position is shewn at Fig. 222. There is some difference in the grip of the file upon various materials, it being greatest on wrought iron or steel, and least on cast iron or brass, so that a file may best be used when new upon brass, then on cast iron, and finally on wrought iron or steel, for it will grip the latter when worn on the former; but the reverse method would not be feasible. During filing the surface should be constantly tested with straight edge, and when finishing, a hand surface plate, being slightly greased with oil and red ochre, will, on application to the work, at once indicate the parts to be taken down. The skin of a casting should always be removed, either by chipping or by pickling in dilute acid, before applying the file, otherwise the teeth would be at once dulled by such a hard surface.

Scraping.—If the surface is to be further trued, recourse is had to the scraper. We will assume that the tool B, Fig. 203, is to be used. It is held in the hand, as shewn in Fig. 223, and the portions to be removed are discovered by smearing a hand surface plate with oil and red ochre and applying the plate to the work. Patches of colour will be transferred to the higher portions of the surface, and when these have been scraped down the work is cleaned again and once more tried, when the colour patches will be found larger in number, but smaller in size and more evenly dispersed. The operation is continued until further accuracy is hindered by the grain of the material. Then we have what is known in the workshop as a *true plane.*

Originating a Surface Plate.—When a new surface plate is required it is generally *copied* from a standard plate kept in the workshop, the method of the last paragraph being employed. But if no such standard be at hand, or if the truth of our first plate be doubted, it is necessary to use three plates in order to originate a true surface. These three plates are first planed truly

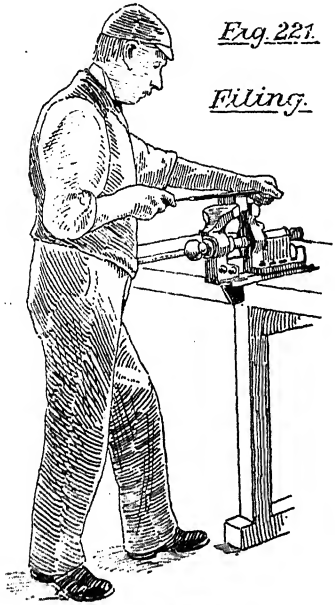


Fig. 221.

Filing.

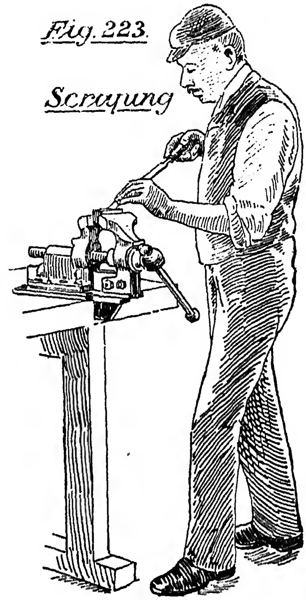


Fig. 223.

Scraping

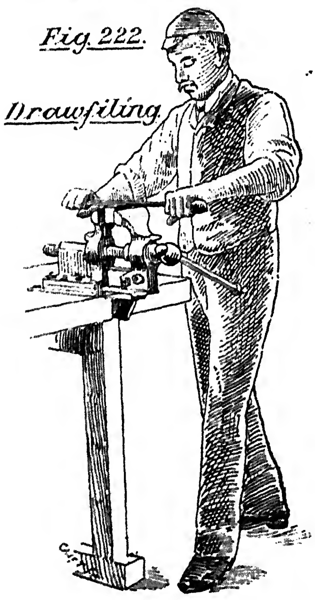


Fig. 222.

Drawfiling.

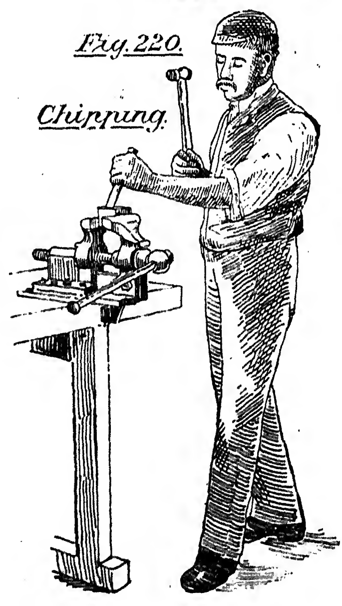


Fig. 220.

Chipping.

by machine, and next filed with a smooth file to obliterate the tool marks. We will indicate the plates by the numbers in Fig. 223*a*. First (1) and (2) are scraped and tried by the colour-patch method, then (2) and (3), and, finally (3) and (1), the cycle of operation being repeated until all fit together with great accuracy. The reason for this method is shewn in the diagram. Thus—(1) and (2) may happen to be convex and concave; then (2) and (3) would be made concave and convex. But if (3) and (1) be now put together, the convexity (or concavity) of both will be apparent, and may, of course, be corrected. But when all three fit equally well they must clearly be equally true.

Although fitting processes are less performed now than heretofore, yet all the best work is trued up by the last-described methods, after it comes off the machine, for however perfect the latter may be, there is always some little distortion caused by clamping the work, which, though slight, must be removed if great accuracy be required. (*See Appendix II., p.* 814.)

Cutting a Screw in the Lathe.—This cannot be fully discussed until velocity ratio of toothed gearing has been entered on, but the practical considerations may be detailed. It will be clear, from what has been said in Chapter V., that if the leading screw be connected to the mandrel in such a way as to revolve at the same rate, a tool of the shape shewn in Fig. 224 will cut a screw groove on the spindle that has been centred in the lathe, of the same pitch as the leading screw thread. If, on the other hand, the mandrel were to rotate at twice the velocity of the leading screw, a screw of half the pitch would be formed on the work, or of twice the number of threads per inch. Summing up then, the pitch obtained will depend on the relative velocities of mandrel and leading screw, a proportionately quicker speed of mandrel giving a finer thread, and a slower speed a coarser thread. The consideration of the proper change wheels to be introduced will be left for Part II., but we may here point out that when both shafts turn in the same direction the screw produced will be right-handed (viz., same as its leading screw), and when revolving oppositewise a left-handed screw will be the result. (*See p.* 484.)

The correct section of V thread, as adopted by Sir Joseph Whitworth, is shewn at Fig. 225, one-sixth of the theoretical

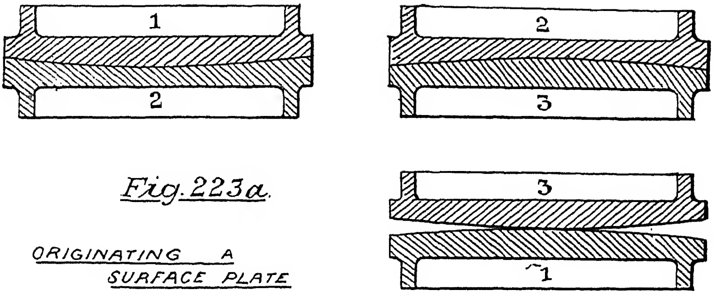


Fig. 223a.

ORIGINATING A SURFACE PLATE

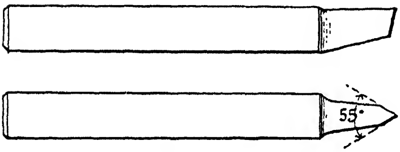


Fig. 224.

LATHE SCREWING-TOOL

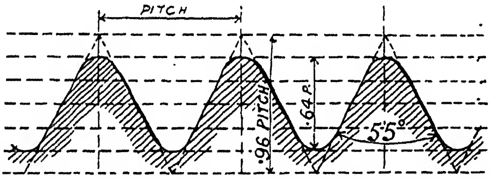
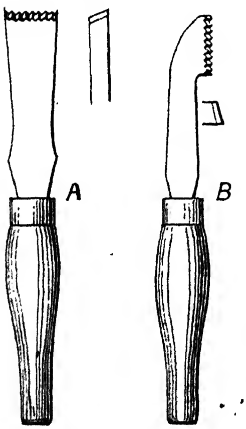


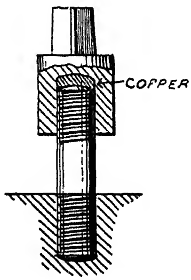
Fig. 225.

SECTION OF WHITWORTH V THREAD



HAND CHASING TOOLS

Fig. 226.



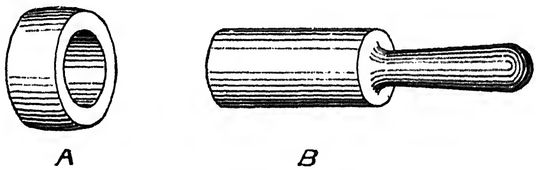
STUD BOX

Fig. 227.

depth being rounded off at top and at bottom, and the angle being 55°. The rounding at the bottom is given by the tool in Fig. 224, but that at the top, as well as the general finish, is obtained by hand-chasing tools. These are seen at Fig. 226, where A is for the spindle, and B is for chasing the nut; the first being held transversely and the second longitudinally. They are both carefully cut to correct section of thread. (*See App. II., p.* 815.)

Fixing a Stud.—Studs are used in places where bolts are inadmissible, because the material cannot be drilled right through. The stud hole being drilled and tapped, and the stud having been turned and screwed so as to fit *tightly* in the stud hole, the former is entered, and a stud box placed upon the opposite end, as in Fig. 227. Outwardly this tool has the appearance of the box key described on page 113; but is screwed internally to fit the stud, and has a small plate of copper at the bottom of the socket to avoid damaging the work. A wrench being applied to the square, the whole is advanced until stopped by the plain portion on the stud, when the box may be removed by a sharp back turn.

Cylindrical Gauges are of great value in securing accurate work. They are shewn at Fig. 227*a*. B being termed a 'plug,'



Cylindrical Gauges.

Fig. 227a.

and A a 'ring' gauge. The first is used for testing the accuracy of a socket, and the second that of a pin, and both are made to such perfection that the tested pin would be found to fit in its socket freely, but with no appreciable shake. There are cases where the ring gauge cannot be applied, and then the 'horseshoe'

form is used instead (*p.* 750), combining both internal and external gauge. For interchangeable work *high* and *low* gauges are required, varying in size by a very slight but known amount, and the aim is to make the work lie somewhere between the two, so that any pair of parts will then fit, and the 'play' between them never be more or less than certain fixed values. (*See p.* 277 ; *also Appendix I., p.* 750.)

Details of Horizontal Engine.—Having fully described machines, tools, and general operations, we shall now proceed to apply the information obtained to enable us to take piece by piece the various parts constituting a 20 *Horse-power Non-condensing Engine*, with automatic expansion gear; and, having received such parts in the rough condition from the Smith or Moulder, to follow them through their various stages, until put together by the Erector to form the complete work. That course has been thought advisable in dealing with this, the most important chapter in Part I, in order to avoid any risk of omitting a good example; it being supposed that if a student could thoroughly discuss the whole of this machine he might be considered reasonably capable of thinking out any new case that might be placed before him. In order to avoid repetition we will make a few premises.

The Marker-off either chalks or white-washes his work before commencing, and obtains the height for his scriber point by first marking the same on the block B, Fig. 192, and then setting the point to this mark. He should know something of the allowances made by Smith and Pattern-maker, which are usually $\frac{1}{8}$ in. all over machined surfaces, and in extreme cases $\frac{1}{4}$ in. Bed plates, for example, warp $\frac{1}{4}$ in. or even more, and special material must be left on their seatings.

Machining is marked on drawings to indicate *all* tooled surfaces; being shewn by red lines; but in our case a thick dotted line will serve the same purpose, thus: ▬ ▬ ▬ ▬. Further, although such drawings are copiously and fully supplied with dimensions, these will be omitted in our examples, the scale being given instead. The sizes represented on the drawing are known as 'finished sizes,' and the allowance on machined parts is left to the judgment of the Pattern-maker or Smith.

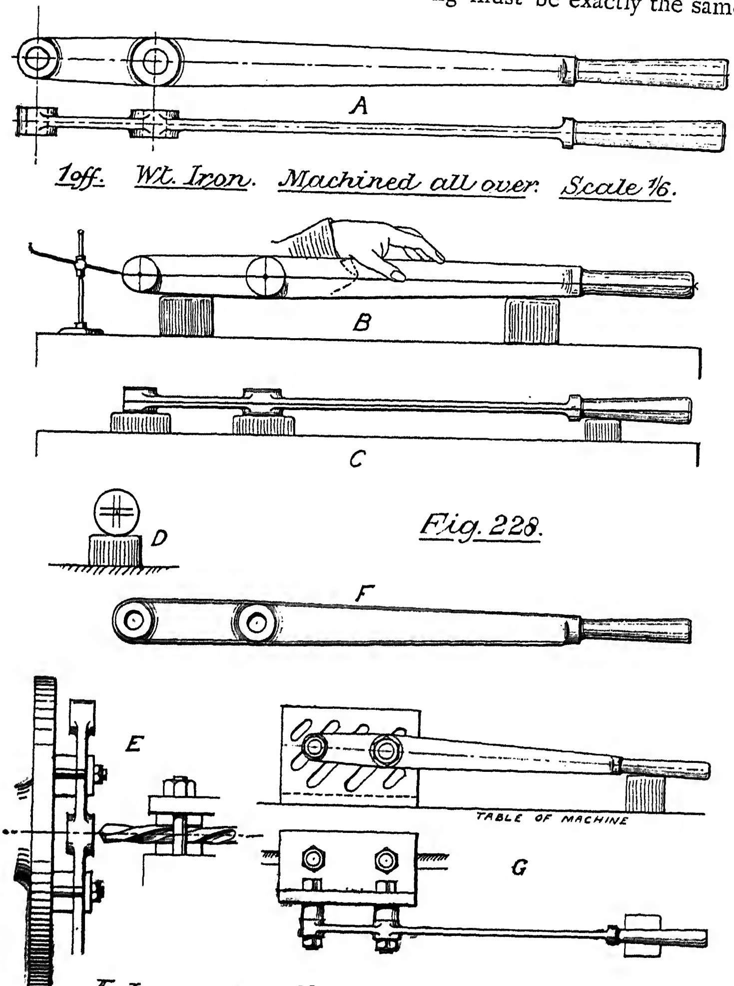
In drilling, there are at least three various sizes that a hole

may be made, although all figured the same on the drawings. *Clearance* size is for bolt holes, so that the bolt may drop in freely; *tapping* size is that at the bottom of the screw-thread; and *gauge* size is divided into 'working fit' and 'driving fit,' the first having both pairs made to gauge, and the second having its socket to gauge and the pin callipered to suit the plug gauge.

As regards the drawings; these are classified as previously described, but we shall further give each article a number in Roman letters; and in nearly all cases the drawing itself will be indicated by the letter A, while the various operations take the succeeding letters of the alphabet. At the close of the descriptions a 'general arrangement' or complete drawing of the engine will be given, and we shall thus have followed in nearly every particular the practical methods of the workshop. One sheet is omitted, that representing a collection of all the bolts and studs to be used on the engine. This has been thought unnecessary, as the capstan lathe has already been described where these parts are tooled. It may further be mentioned that there are always more ways than one of performing the various operations, both as regards sequence and the tools used, and it may also follow that each method is equally good; in many cases, too, where the marking-table is mentioned in our descriptions, it might be found more convenient to scribe the work while in the machine.

I. Regulator Lever (Fig. 228).—This must first be set upon edge, on blocks, as at B, until level; and a centre line be scribed all round it. The circles may be struck, just to see if the stuff 'holds up,' and the length of the handle marked off from these. Now punch all the five centres. (A method of centering with scribing block by laying the lever successively on its four sides and scribing any convenient height is shewn at D.) Lay the lever next on its side (C), and pack up until the centres are quite horizontal, as measured with scriber. Then scribe the centre line all round, and mark at the same time the thickness of the bosses, and of the lever itself, as measured from this centre. Next put in the lathe, and square-centre; then turn and polish the handle. Remove and clamp to the table of a shaping machine, so as to shape across the flat parts; then clamp on the lathe face plate as shewn at E, for the purpose of drilling the

bosses and turning them. Of course the boss must be carefully centred on the plate, and the blocking must be exactly the same

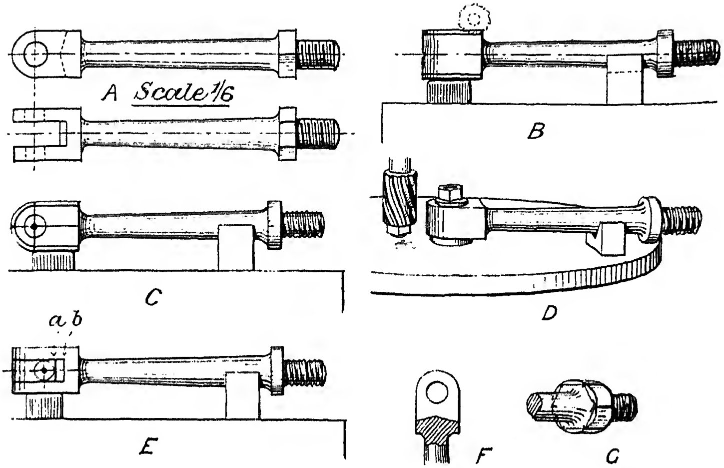


I. Lever for Regulator Valve.

thickness. The drill point is placed against the centre of the boss, and the loose headstock brought up to the other end of the

drill. The latter is then packed in the slide rest, and firmly grasped, when the drilling may proceed. The bosses are surfaced to the scribed mark, and to the shaped lever; the diameter being gauged by callipers. When these are finished, the sides are marked out as at F, and the lever next clamped on an 'angle plate' placed on the table of a planing machine (see G), being packed at such an inclination that the edge may be planed; and four settings are of course necessary. The angle plate is an appliance which will be found useful for a variety of purposes. Now finish off the lever by draw-filing and emery cloth. If the work be too long to allow the bosses to be turned, the latter are tooled as separate pieces, having a portion of the lever attached, and are afterwards welded to the handle by the smith.

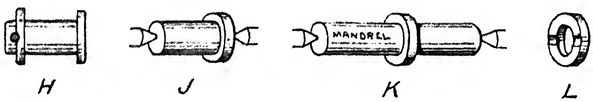
II. **Brackets for Regulator Lever** (Fig. 229).—First centre at the ends as at B, and punch; then try in the lathe to see if there is sufficient stuff at the middle. Turn the shank to dimensions, gauging with callipers, and cut the screw thread in the lathe at the same time; the taper of the shank being obtained by setting the top slide of the rest by the requisite amount, and giving a hand feed to the tool. Polish while in the machine, with file and emery, all but the collar, which may be left rough, because it is to be afterwards cut; the diameter then being made equal to that across the corners of the hexagon. Now remove from the lathe, and, setting again on the table in the position B, line out the flat cheeks of the fork, and shape or mill these. Upon the tooled surface thus obtained further lining is performed as shewn at C, the centres being again placed exactly horizontal. Strike the pin hole and punch. Drill the hole in machine vice to gauge, and, bolting down to the centre of a slotting or vertical-milling table D, tool all round with hand and machine feed. Once more line out, this time for the fork slot as at E, and also mark a circle for drilling, making sure that the line *a* is taken for this, not *b*. Drill the hole last marked, and take out the rest of the fork slot in the slotting machine, finishing by cutting the oblique portion in the vice by chipping and filing (F). It may here be mentioned that all bright work is held in the vice between plates of lead resting on the jaws, and called 'vice clams.' Lastly, cut the hexagon on the collar by dividing out as at G, and filing off the flats.



II. Bracket for Regulator Lever. 1 off. W.I.

Machined all over

Fig. 229



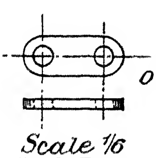
III Pins for Regulator Lever 3 off. W.I.

various sizes.

Fig. 230.

PARALLEL RIMER OR BROACH.

Fig. 231



IV Links for Regulator Lever.

2 off. W.I.

Fig. 232

III. Pins and Washers for Regulator Lever (Fi 230).—Three of these are required, of various sizes, to be mac to the drawing H. Centre for the lathe ; turn to gauge and poli: as at J. The washer is made from a piece of plate, by fir drilling the hole, and afterwards turning the rim on a mandrel, shewn at K. Then, the lever, bracket, valve spindle, and linl are all fitted together; a *broach* or *parallel rimer* (Fig. 231), exact gauge passed through each set of holes to clear out tl irregularities produced in drilling, the pins put into place, an the split pins marked off —— a groove being cut in the washer at L, to prevent turning and undue wear.

It is advisable to make all pins of steel that have to withstan much wear, and their corresponding lever bosses, if of wrougl iron or mild steel, should be case-hardened.

IV. Links for Regulator Lever (Fig. 232) are marke on a piece of plate as at M, which has first been planed on a four sides, then drilled, cut in two pieces, and bolted togethe They are finished off by filing in the vice, though, if large, the would be slotted round, or milled. Polish with emery.

V. Regulator Valve Spindle (Fig. 233).—Lay this o its side, in V blocks, as at B; centre the ends, and scribe th flats. Then put in lathe and turn to exact diameter, at the sam time cutting the screw. Remove, and tool the flats in a shapin machine. Now mark off the eye, as at C, and punch the centre drill the hole to gauge, and take off the outer material with vertic milling cutter fitting the curve *a*. Finish off in vice and polish.

VI. Nut for Valve Spindle (Fig. 234).—Lay on tabl and line out for thickness, as at B; plane or shape the flats; mar off the hole, as at C; and, placing the nut in a concentric chucl bore and screw in the lathe as at D, so as to fit the valve spindl easily.

VII. Regulator Valve (Fig. 235).—After cleaning wit rough file to remove fins, this has only to be machined on certai surfaces, as shewn by thick dotted lines on the drawing A. A the face must be reasonably true with the lugs, first find centre c the latter, as at B, and square a line from the back surface *a a* having previously blocked the hole with a piece of hard wood do this for both lugs. To produce this centre line on to th

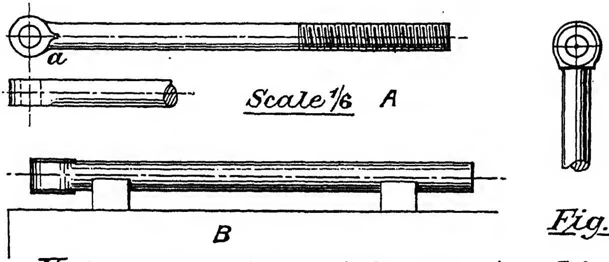


Fig. 233.

V. Regulator Valve Spindle 1 off. W.I.

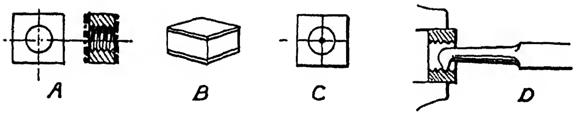


Fig. 234.

VI. Nut for Valve Spindle 1 off. Gun Metal.

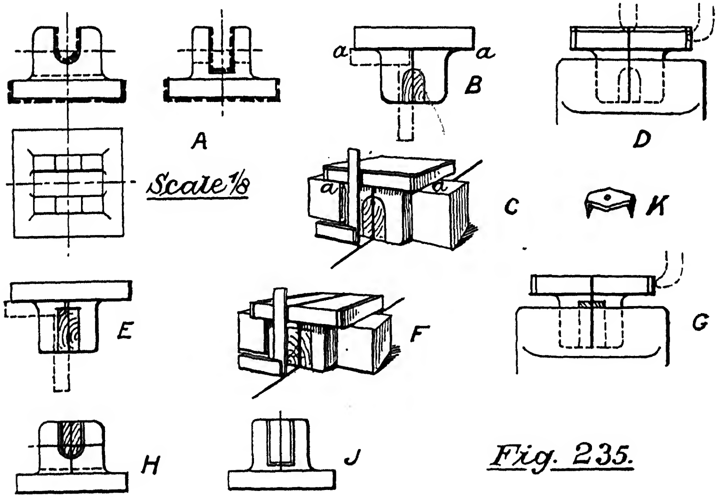


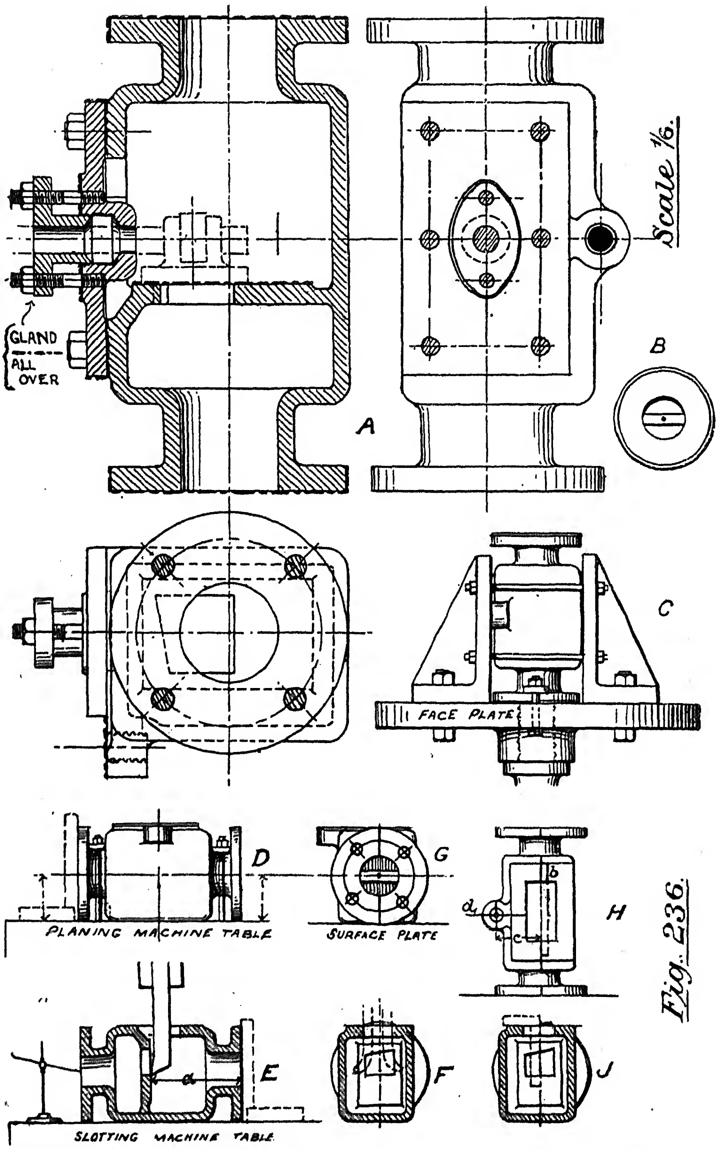
Fig. 235.

VII. Regulator Valve. 1 off. Gun Metal.

edge of the plate, the valve is supported, as at C, on the marking-table, so that its back surface, our present guide, is level, and the

just scribed vertical line put in contact, both at back and front, with a line which has been marked on the table. Then set up this line with square, as shewn, so as to mark the centre of the valve plate, and measure off to right and left the width of the valve. Scribe also the thickness of the plate all round. Now set in machine vice to plane the top surface, as at D, with a front tool, and the edges with a side tool, and be sure that the travel of the tool is exactly coincident with the scribed lines. The valve is now removed, and treated similarly for planing at right angles to the former direction. For this the fork is blocked, as at E, and the centre squared; next produced upward, as at F, and the width of valve marked, then planed, as at G. There only now remains the cutting out of the fork, which is lined by squaring and scribing, as at H and J; then J is planed out, and H is finished by hand. Finally scrape the valve surface very truly, as described in a previous paragraph. It should be mentioned that when wood is used to block or bridge a hole, and a centre required, it is advisable to shape a small piece of tin or zinc, as shewn at K, to receive the compass-point.

VIII. Regulator Valve-box, Cover, and Gland (Fig. 236).—Commence by bridging or 'spanning' the two end holes, and striking the circle representing the diameter of the flanges, as shewn at B; measure also the length of the box over the flanges, and mark this. The valve-box is now to be mounted on the face plate of a lathe, and as the casting is rather long, it must be supported by angle plates, as shewn at C, being tightly bolted between them, as well as having one flange fastened to the face plate. Having been carefully adjusted until central, it is turned on one flange and surfaced; reversed, and turned on the opposite flange. Next place the box on a planing machine, as at D, making sure it is both level and square, packing if necessary, and having scribed the top seating and boss, measuring from flange centre, plane these. Remove, and bolt to slotting machine in a similar manner, as at E, and slot the front face, measuring the distance *a* in finishing. It should be noticed that two tools are here necessary, cranked respectively to the right and left hand, as at F. Set out the bolt-holes in circular flange as at G, and drill with clearance drill. Set level as at G, and, squaring up the centre line, join this



Scale 1/6.

Fig. 236.

VIII Regulator Valve Box. 1 off. Cast Iron.

along the top, as at H Then, in position H, scribe and square the centre *d*, measuring the distance *c* from a straight edge at *b*. Strike the circle for the hole, drill this tapping size, and tap to suit the Bracket No. II. To ensure rigidity, the bracket should have been screwed rather 'full,' and be now taken down with stock and dies until a perfect fit in *d*. The port is to be marked off, as at J, with square and straight edge, and measurements from the square flange, and the edges are then to be chipped and filed by hand, an operation involving some trouble.

The **Cover** is to be planed and drilled. Find the centre of the gland seating as at B, Fig. 237, and mark also the centres at each end; then draw a line across. Set the cover level on the marking-table, as at C, and squaring up the centre *d*, scribe the thickness of the plate, and mark its width. Set central, on the planing machine, in a machine vice, or its equivalent, as at D and E. Level the cover to the scribed lines, and plane the side surfaces, *a a* and *b b*, as well as the edges. Remove, and square across for the adjacent sides, as shewn at F, using the gland seating as a guide. Then set in the planing machine, and tool the edges; finish the surfaces *e e* to the same level as *a* and *b*. Now reverse the plate, and, setting level in the machine vice, as at G, scribe the gland surface to measure correctly from the surface *a b e*, and plane. The cover is next to be marked for drilling, which is shewn at H, and the holes *gg* drilled to clearance size, *h* to tapping size, while *j* must first be drilled for the smaller diameter, and the stuffing-box afterwards taken out with a pin drill specially cut, as at J.

The **Gland** is first drilled in the lathe (B, Fig. 238), which may be done more truly by blocking up the hole with wood through its entire length, and letting the drill take this out. The front may be surfaced at the same time. Next place upon a mandrel as at C, and turn down to dimensions. Remove, span the centre, and mark off the gland face as at D; then drill the holes (clearance), and finish off the edge with dead smooth file and fine emery. The cover holes are lastly to be marked off on the box by tracing through from the cover, then drilled and tapped.

IX. Valve Spindles: Main and Expansion (Fig. 239).

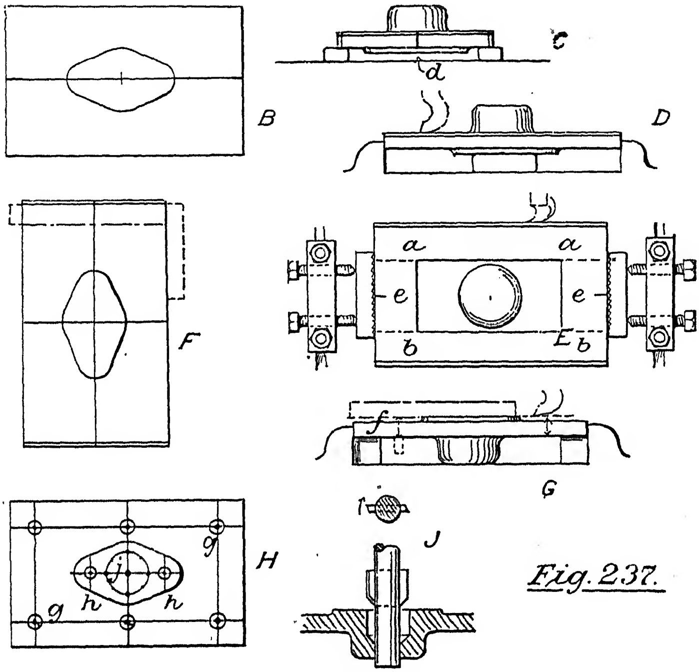
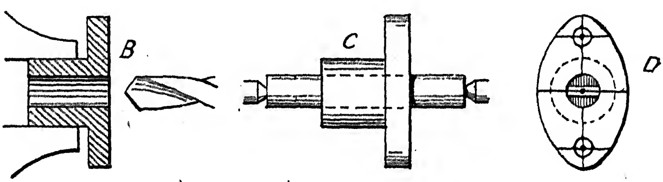


Fig. 237.

Cover for Regulator Valve Box.

1 off Cast Iron.



Gland for Valve Box. 1 off.

Gun Metal Fig. 238.

VIII a.

As with No. II., these are first centred, and scribed on the flat cheeks (D); then turned and screwed, and shaped on the cheeks. The hole is next struck as at E, drilled, and the outer curve milled; and the fork (F) is taken out last by drilling and slotting. Broach right through, and turn the pin as was described for No. III.

The **Nuts** are best finished by putting a number of them after drilling to tapping size, upon a mandrel, which is then placed between dividing centres on a milling machine, and milled by means of twin mills (see B). They are to be turned axially through 60° at each operation, and must be afterwards tapped, and chucked in the lathe for facing and chamfering.

X. Expansion Eccentric Rod (Fig. 240).—Centre this; also mark the length between the shoulders, and square up the thickness of the T end. Turn to the requisite taper by 'setting over' the loose headstock, as shewn in plan at J, so that the front surface of the rod will then be parallel with the lathe bed. The amount of set-over will, of course, be equal to half the difference of the two end diameters. Surface also the T end. Remove from the lathe, lay level as at K, and scribe the cheeks *b*. Square and scribe the tee at *a* to dimensions, measuring from centre, and strike also the bolt-holes. Drill these to clearance size, and shape *a* and *b*. Then mark out the eye as at L and mill this with a cutter having the proper curvatures. The rod is long, but as the milling only requires it to sweep through a semicircle, there will be no serious difficulty if it be well clamped.

XI. Main Eccentric Rod (Fig. 241) presents no difficulty after the previous descriptions. (*See Appendix II., p.* 818.)

XII. Intermediate Valve Rod (Fig. 242).—This also would be tooled by previous methods. The manner of fastening the pin is worthy of notice. The bearing surfaces of the fork are but narrow, and it is unwise to allow movement at that place; the die, on the contrary, has a good wide surface, so it is there only that wear should be allowed. After the pin is put in position, a parallel hole is drilled right through the fork, and enlarged with taper rimer, the pin for this hole being turned in the lathe with an oblique hand feed. *All these pins are of steel, and all wearing surfaces are case-hardened.*

The **Die** is surfaced and bored in the lathe, and afterwards

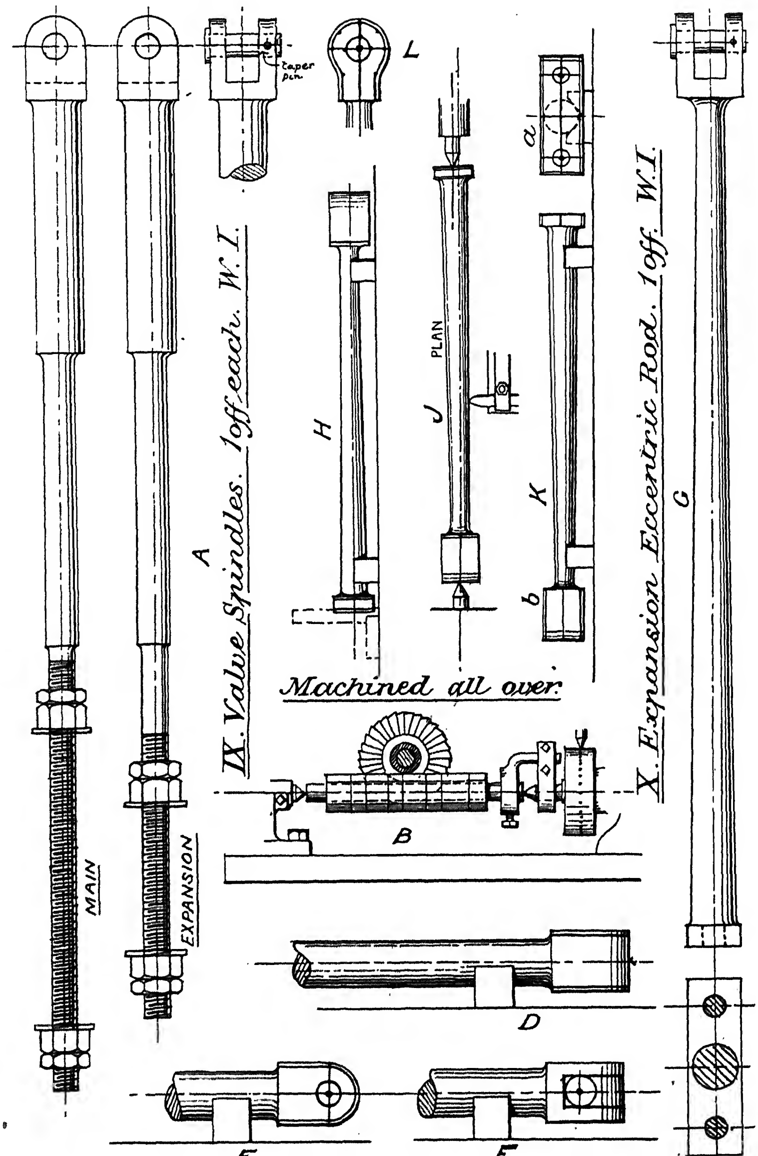


Fig. 239. Scale 1/6. Fig. 240.

shaped to dimensions, leaving sufficient excess of width to allow of accurate fitting to the Radius Link, after which it is case-hardened and polished.

XIII. Guide Bracket for Valve Rods (Fig. 243).—The machining is shewn on the drawing at A. Set the bracket vertical by trial with square as at B, and line out the base *a*. Scribe also the line *b* all round the casting, and at the proper height from *a*. Lay level on its side as at C; find by measurement the height of the boss and that of the foot centre. Scribe the thickness of boss. Next shape to these lines as at D, the boss with a front tool and the foot with a side tool. It should be noted here that a side tool ought never to be used if it can be avoided, for there is a great twisting action thereby produced which is calculated to wrench the tool from its box; but a good deal is sometimes sacrificed to save two settings on the machine. Re-scribe the line *b* from the marks left on the side of the boss, and lay the bracket on its side as at F, packing until the centres are level; then scribe the heights of the large holes on both faces and strike the circles. The casting being hollow, the core-hole must be spanned as at G, in order to strike the bolt-holes, whose centres are found by scribing a horizontal line and squaring a vertical one when in the position C, and then bisecting the right angles obtained. The bolt-holes are drilled as at H, but the large holes are bored in the lathe, the bracket being clamped to the face plate, and the latter provided with a balance weight. This will be understood from B, where the face plate will be seen dotted, and the bracket clamped in position for boring *b*. In all such cases it is necessary to first drill a hole large enough to admit the boring tool.

The **Bushes** are bored in a chuck, and finished on a mandrel, and afterwards driven into the bracket, a block of wood being placed upon them to receive the blow of the hammer.

The oilcup cover is drilled for the hinge-pin, and finished by hand, and the oil-holes drilled, and countersunk slightly at the top. An $\frac{1}{8}''$ spiral channel should be chipped in each bush with round-nosed chisel to allow the oil to flow.

XIV. Eccentric Sheave and Straps (Fig. 244).—The cast-iron sheave will be taken first. It is of the solid form, being

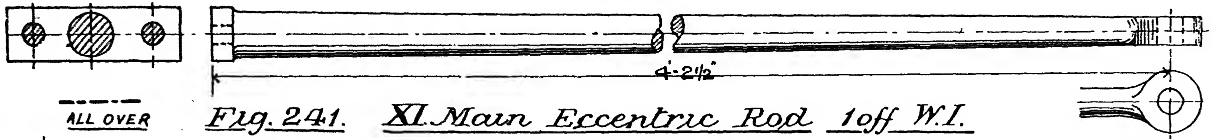


Fig. 241. XI. Main Eccentric Rod 1 off W.I.

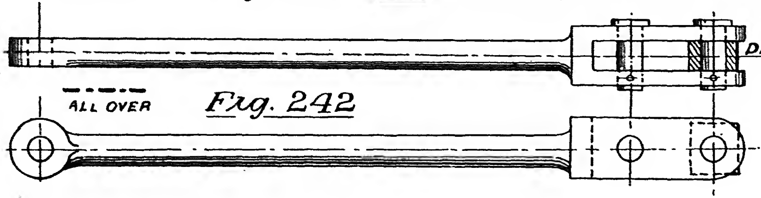
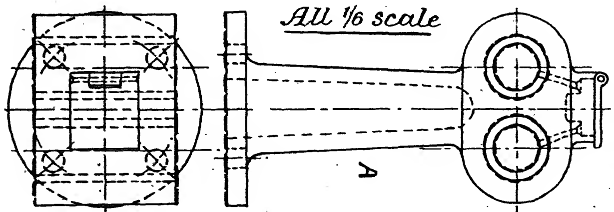
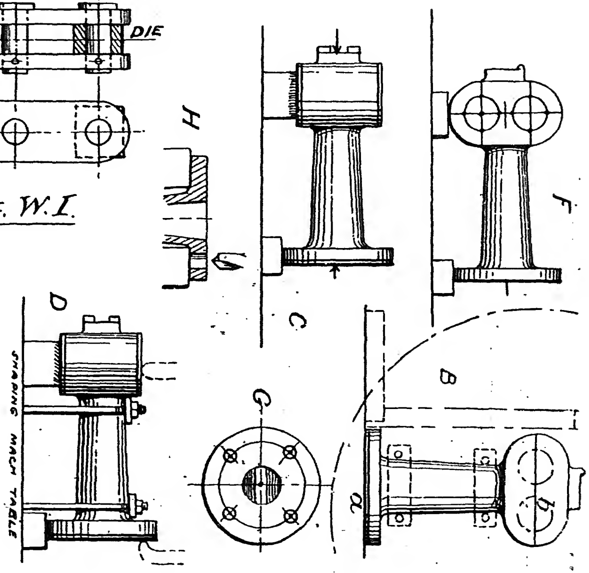


Fig. 242

XII. Intermediate Valve Rod 1 off. W.I.





All 1/6 scale

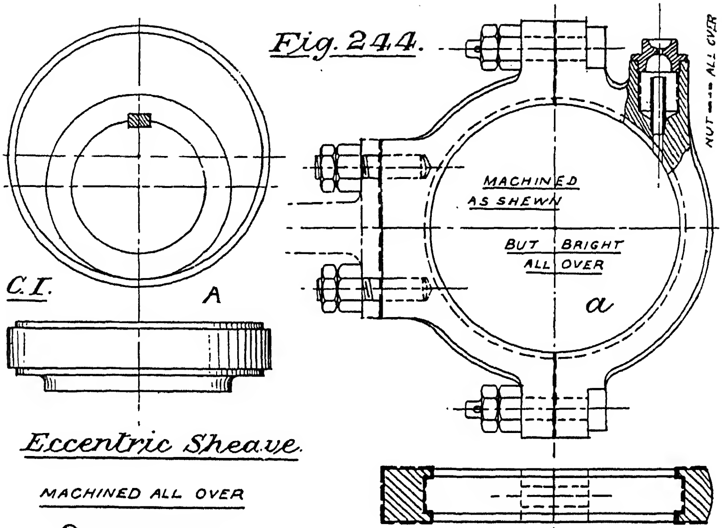
Fig. 243

XIII. Guide Bracket for Valve Rods

slipped on to the shaft lengthwise. But there are many cases where it is necessary that the sheave should be in halves for this purpose, and the machining would be then performed in a very similar manner to the eccentric straps to be described, namely, by bolting together the halves before turning. The drawing of the sheave is shewn at A. Lay the casting level on the marking-table, as at A_1, and scribe the various thicknesses; span the hole, as at B, and strike a circle for its diameter. Grip in the dog-chuck, as at C, bore, and surface the projecting boss and the face of the sheave, marking the diameter of the boss in the lathe. As the sheave has to be chucked eccentrically, the face plate must, of course, be balanced. Next reverse, and turn the opposite face of the sheave, this time chucking centrally, as at D, and setting the already turned boss close to the face plate. Lastly, the outer circle is struck out, as at E, by re-bridging the centre, and marking the exact eccentricity on the centre line at x, and the work is then bolted to the face plate, as shewn at F, each portion of the rim being measured in position, and carefully turned exactly to dimension, because it must be a correct 'working fit' with the strap. The key-way may be slotted out. (*See App. I., p.* 751.)

The **Straps** (drawn at *a*, Fig. 244) are first marked off, as at A_{11} and B_1 (Fig. 245), with the proper allowance for machining the feet, and the two are then bolted down together to the planing table, as at E_1. The bolt holes are next scribed and squared, as at D_1, the casting lying level on its side, and these are drilled, as at E_1. The thickness of the feet for the front strap being lined at F_1, and the stop for the bolt-head at H, these are cut out, the first with pin drill, as at G (known as 'knifing' or 'face-arboring'), and the second with chisel and file. Now place face to face in the vice, and broach the bolt-holes right through; then, having turned the bolts to a good fit, fasten both straps together. Lay the bolted straps level, as at J, scribe the width, and grip in a dogchuck, as at K, to face both sides, setting carefully for each. Leaving the work in the chuck, examine the outer rim for centrality (for this cannot afterwards be turned), and mark off the inner circle for boring, measuring with callipers and rule as the work proceeds. Remove from the chuck, and scribe the remaining surfaces—*b c d*, as at M, measuring from the turned

Fig. 244.



Eccentric Sheave.

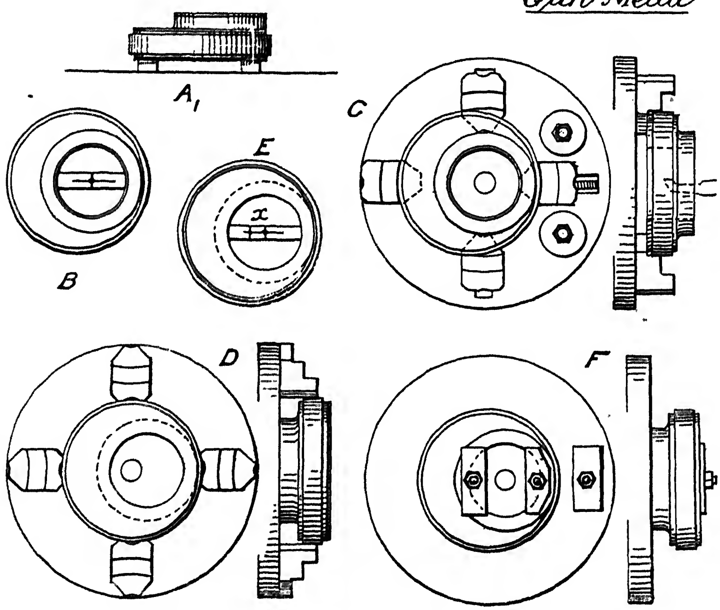
MACHINED ALL OVER

2 off each.

Scale 1/6

XIV. Eccentric Straps

Gun Metal



[illegible] after which [illegible]

The Oilcup Cover Nut [illegible]

XV. Slide Bars are [illegible] Fig. [illegible] and the [illegible] are then [illegible] the [illegible] The width of the groove [illegible] depth, finishing the surface with a [illegible] tool, and the corners with a [illegible] tool. The sides of the bars are planed in a similar manner. The ends are shaped [illegible] length and the holes marked off and drilled [illegible] casting, to [illegible] and are [illegible] to be cleaned out by hand. After [illegible] the groove should be scraped to suit the slide block.

XVI. Slide Bar Bracket and Distance Piece (Fig. [illegible]). The bracket is placed upright on the marking table, as at c, and set vertically by trial with square on both sides, measuring as at a a. Scribe the thickness of foot, and place upside down on the planing machine, clamping in machine vice; then plane the foot. Now reverse the bracket, and, clamping right side up, mark off the height of the bosses, measuring from the foot, and plane these. Remove, and block up on marking table, as at c and d, and scribe a centre line all round; then measure the positions of the stud-holes, so as to agree with regard to the square bosses. The bolt-holes are to be marked off by setting the casting as at e, and measuring with a square the two dimensions shown, the difference of which will be the distance between stud and bolt centres. Punch and drill the bolt-holes to clearance, and the stud holes to tapping size. Finally, tap the latter.

The Distance Piece is a simple example of shaping, which done, the hole is marked off and drilled to gauge.

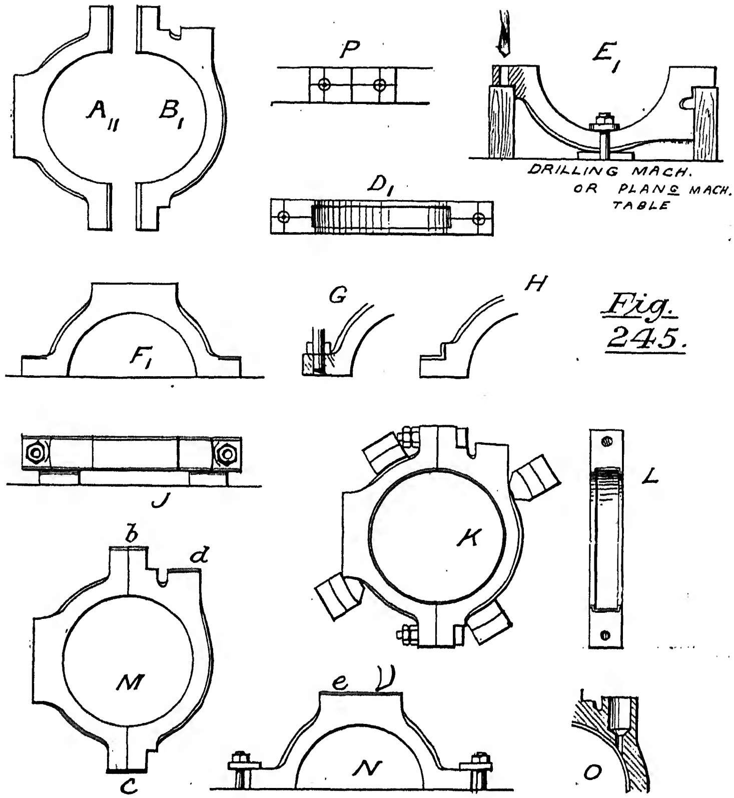
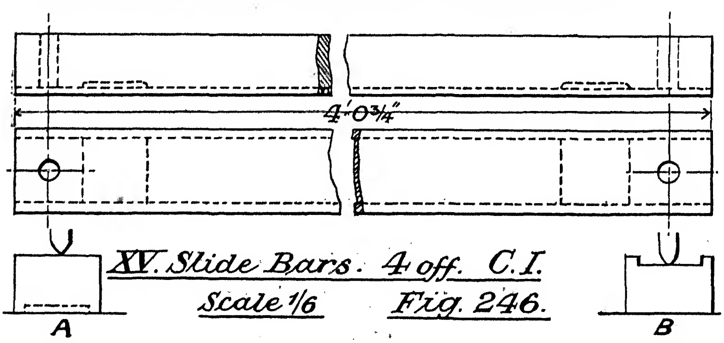


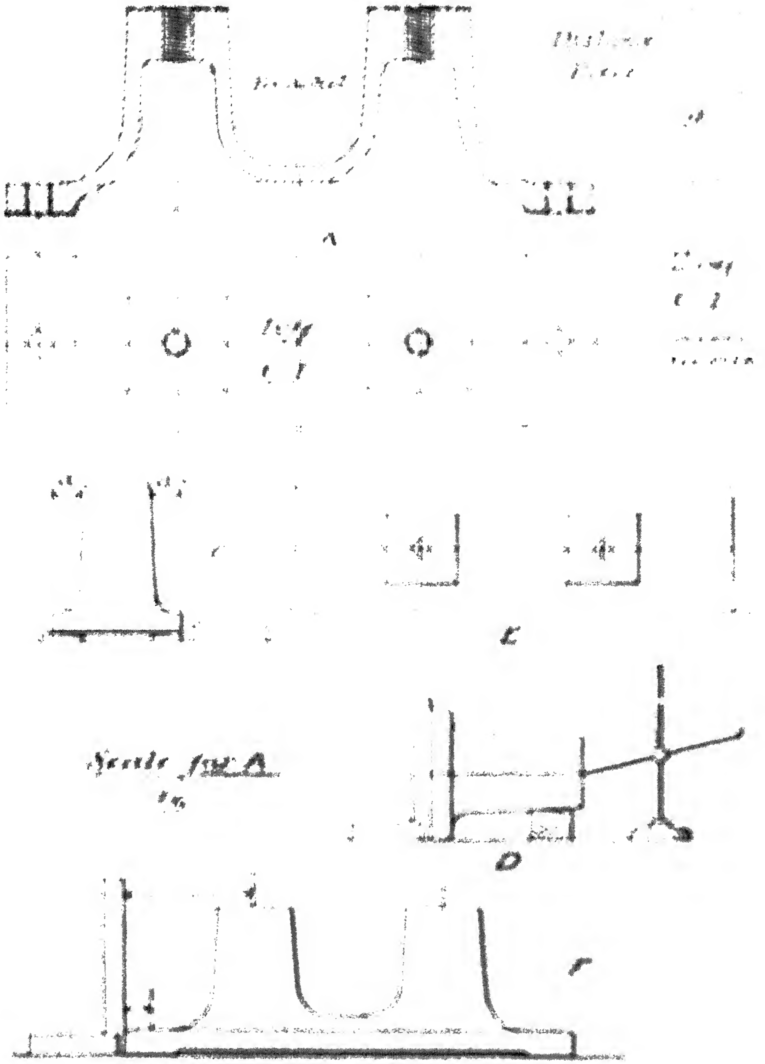
Fig. 245.

XIVa Eccentric Straps. (continued)



XV. Slide Bars. 4 off. C.I.

Scale 1/6 Fig. 246.



XVI Bracket and Distance Piece for Slide Bars

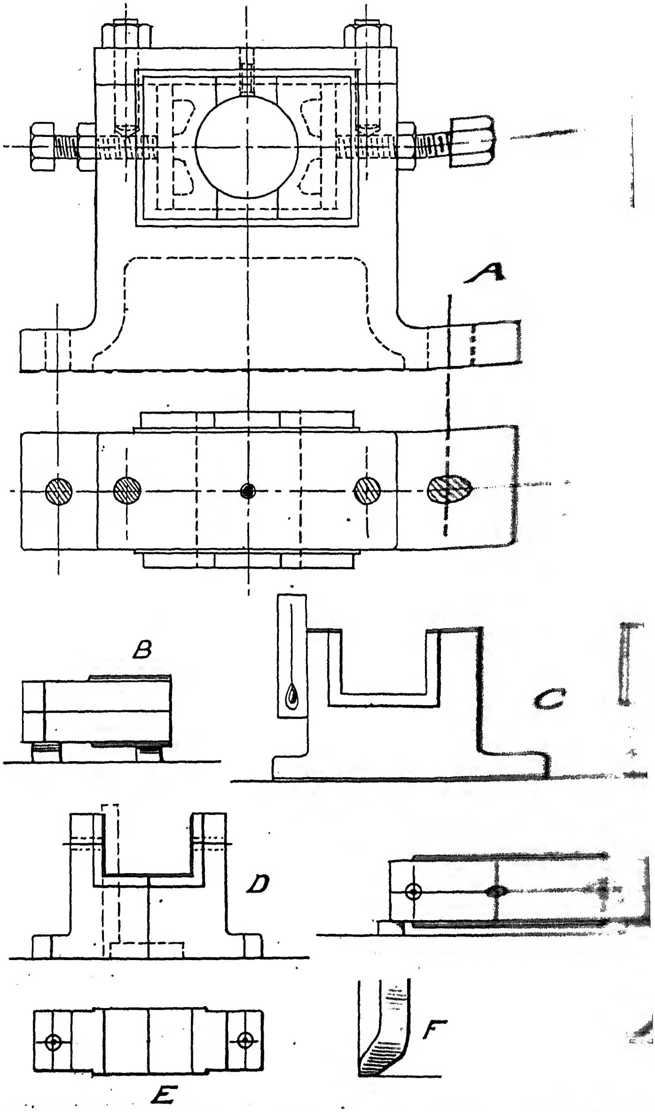
Fig. 247

XVII. Crank Shaft Bearing (Fig. 248). The bearing is first laid level on its side as at B, the centre line obtained, and scribed round. The seatings for the brasses are measured and also scribed, after which the bearing is set up as at C (see both views), and adjusted by line and square till plumb. The foot is next lined, and the top of the bearing taken from this, making sure that there is sufficient stuff left in the bush socket. Now plane in turn the seating sides, the foot bottom, and the top. Stand the casting again on the marking table as at D, and find the centre of the socket. Square this up, as well as the socket sides. Scribe the bottom of the socket, measuring from the foot, and line the bearing centre all round. Square these lines across as in plan at E, and mark them on the opposite side. Find the centre of the set screw-holes, and measure the foot bolt-holes from the vertical centre line. Plane out the bush socket—the sides with a side tool, and the bottom with a front tool, finishing with a flat tool, and the corners with the 'corner' tool shewn at F. Drill the holes.

The **Cap** or 'keep' is set on edge to line the seatings and scribe the two bolt-holes and oil-holes, as at G, being first, however, planed to thickness on its bottom surface. After planing also the seatings, and drilling the holes, it is placed in position on the top of the bearings, and the bolt-holes marked through to the latter. These are next drilled and tapped, and the studs put in place.

The **Brasses** are shewn in Fig. 249. Being first laid on its side, as at A, the large brass has its width marked and its lips lined for thickness, and is then planed. The front and sides are next lined out on all surfaces to dimensions, measuring from the planed surface, and trying for depth of stuff between the lips, the brass being meanwhile packed with sides truly vertical, as at B. The whole is now planed by clamping in the successive positions, C, D, and E, so that every surface is done, either with a side or front or knife tool, the depth of the middle surface being gauged from the lip; and the small brass F is similarly treated. (*See App. I., p.* 752.)

The packing plates are next machined, and all is ready to put together. The brasses and packing are to be carefully smooth-filed and scraped until they bed perfectly into their places in the



XVII. Pedestal for Crank Sh…
2 off. C. Iron. Stens: G…

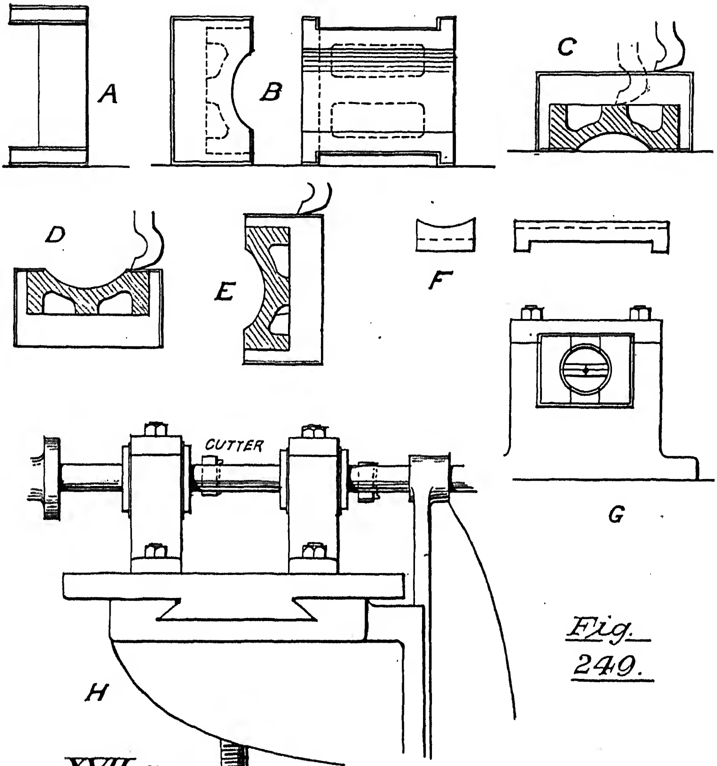
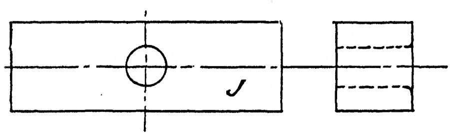


Fig. 249.

XVII a.
Crank Shaft Pedestal. (contd.)

XVII.
Slide Blocks
2 off. C.I.



Scale 1/6. Machined all over.

XIX.
Gudgeon.
1 off W.I.

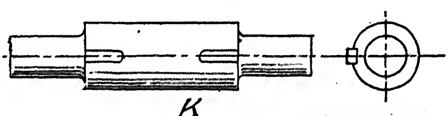


Fig. 250.

[illegible]

XVIII. Slide Blocks. [illegible]

XIX. Gudgeon. [illegible] V [illegible]

XX. Crank Shaft. [illegible] already [illegible] at page [illegible] V blocks [illegible] straight edge [illegible] Wings and countersunk. [illegible] shown at c, bored so as to [illegible] the shaft. Place the latter in the position shown at [illegible] packing and V blocks. A countersunk having been already formed in the wings at a distance *d* equal to the throw of the crank, this has to be adjusted until in line with the crank pin, by measuring from the double pointed line at *a a* and by setting [illegible] the crank web as at [illegible]. Then the set screws are tightened firmly on to the shaft, with packing strips underneath of such thickness as to adjust to the distance *d* with great accuracy. The whole may now be lifted from the blocks so as to admit the [illegible] *b b*, which have their end [illegible] turned till perfectly equal. Lift the crank into the lathe, and place the wings between the centres, when the crank pin will of course be also central. The object of the wings will now be seen, for they either entirely or partly

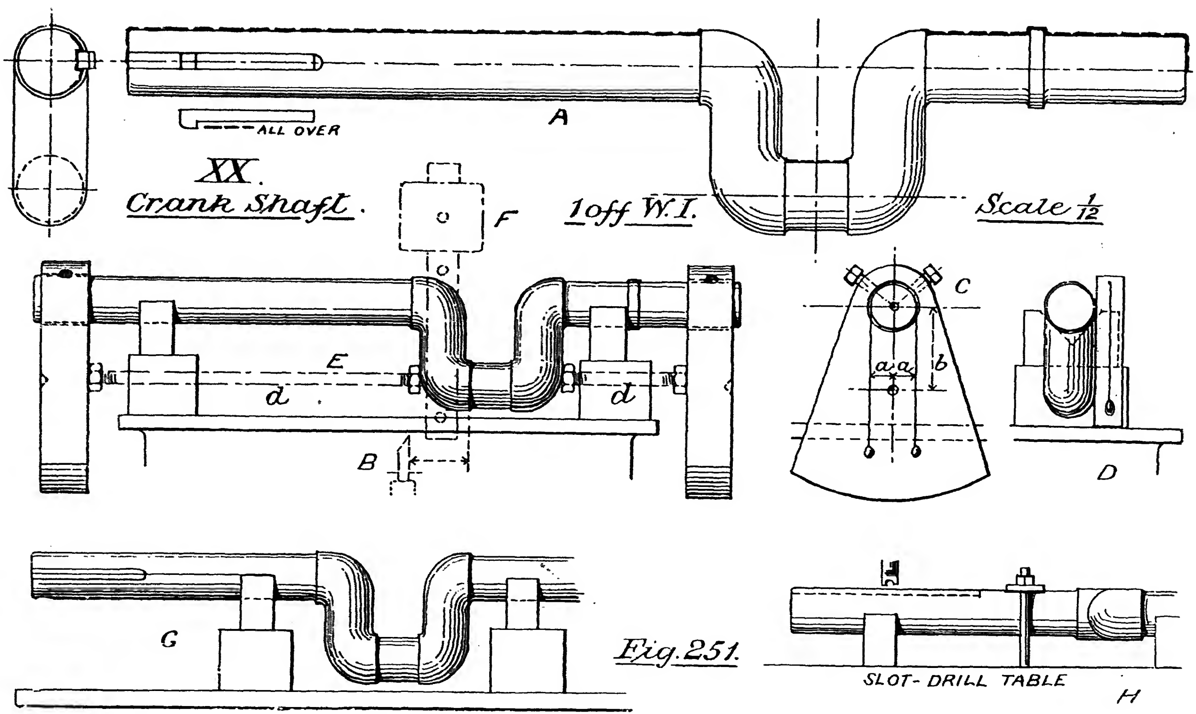

XX.
Crank Shaft.
1 off W.I.
Scale 1/12
ALL OVER
A
B
C
D
E
F
G
H
a a
b
d
d
SLOT-DRILL TABLE

Fig. 251.

balance the [illegible] [illegible]

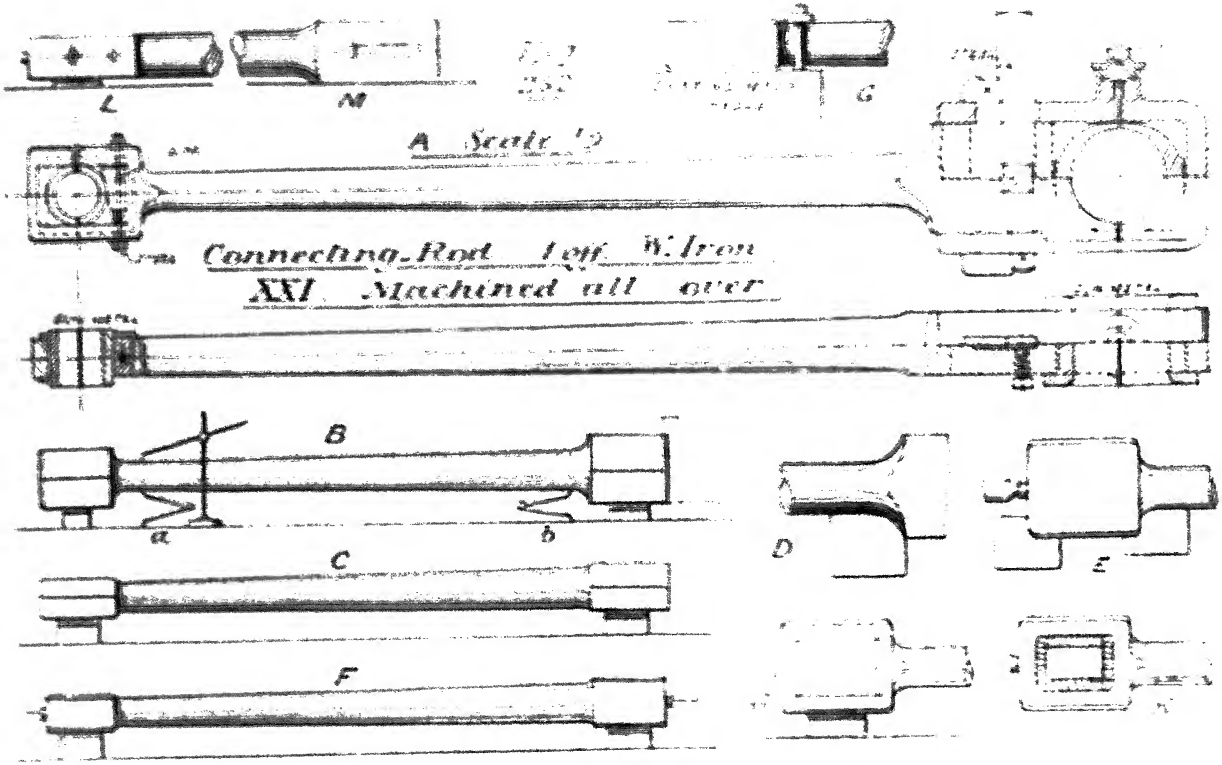
[illegible]

[illegible] The same is done on the other side of the [illegible] and the shaft then turned to diameter, [illegible] with trammels, or [illegible] may be balanced, if [illegible] the web, but [illegible] and balance. The [illegible] are [illegible] the [illegible] and not [illegible] and are not therefore shown.

The key at a is planed from good steel, and a [illegible] in fig., the fitting to wheel being left to the [illegible].

The crank web should be slightly chipped to remove rough scale, filed with rough file, and afterwards painted.

XXI. Connecting Rod (fig. [illegible]). The large end of the rod is supplied with a strap enclosing the "butt," and a gib and cotter for tightening purposes, and the small end is turned "solid." The rod is first taken and laid flatly level on the table, as at a. It must be centred for turning by the following method: Caliper each round end, and let the difference of a and b be half the difference of the diameters. Then "level" and measure both heights at a, add together, and halve. Do the same for b, and so adjust till these quantities be the same. Then scribe this dimension, the height of the centre, all round, as



A Scale 1/2
Connecting-Rod Left W. Iron
XXI Machined all over
L
M
G
B
a
b
C
F
D
E

shewn. Trammel between the shoulders, and square up the vertical lines, as well as the end lines; measured from these. Next lay the rod flat as at C, and scribe the centre line round in a similar manner to the last. Punch centres, and place in lathe, testing with chalk, and square-centreing. Now set the poppet head over by half the diameter difference, and turn the taper portion up to the shoulder radii, as shewn at J, Fig. 240. Set the poppet head true again, and surface the ends of the rod, also the shoulders up to the radii. These last require very careful turning. They are to be roughed out by means of a combination of surfacing and traverse feed, and semi-finished by a broad tool ground to the curve, the position of which is gradually changed by turning round the top rest until the whole curve is gone over piece by piece. The last finish is given by hand with the same tool very sharply ground. Of course the work must be continually tried by means of a sheet iron copy called a 'template,' shewn at D and E, the lathe being stopped at each trial; and the outer curve of the solid end is to be finished in like manner.

Remove from the lathe, finished, but not polished, and lay on the surface plate as at F, packing till level. Scribe the thickness of the butt and solid end, then fasten, as at G, across a shaping machine having two tables, and shape. Similarly also for the depth. Return to the marking table. Scribe the centre line afresh, and plot out the square hole in solid end as at J; do this on both sides, and well dot all round it. This may now be cut out in one of two ways—(1) a hole may be drilled large enough o pass a slotting tool, by twist drill and pin drill, and the rest of the work done by slotting; (2) a probably better method, is to take out all round by means of slot-drilling tool, drilling, say, a quarter of an inch down, traversing all round as at K, then a little further down, and so on till the hole is completely cut, finishing the sides with a milling cutter and the corners with a corner tool. There is then very little work left for the file.

Now mark off the bolt-holes at L, on both sides of the solid end, together with the oil-hole, and the cotter-hole at M. Drill the bolt-hole from each side, broach through, and countersink the oil-hole. Take out the cotter-hole in slot-drilling machine

as above described, until cut right through, when there ought to be little finishing with file. Drill and tap for set screw in butt end, return the rod to the lathe for polishing, chip off the centreing pieces, draw-file, and polish the ends.

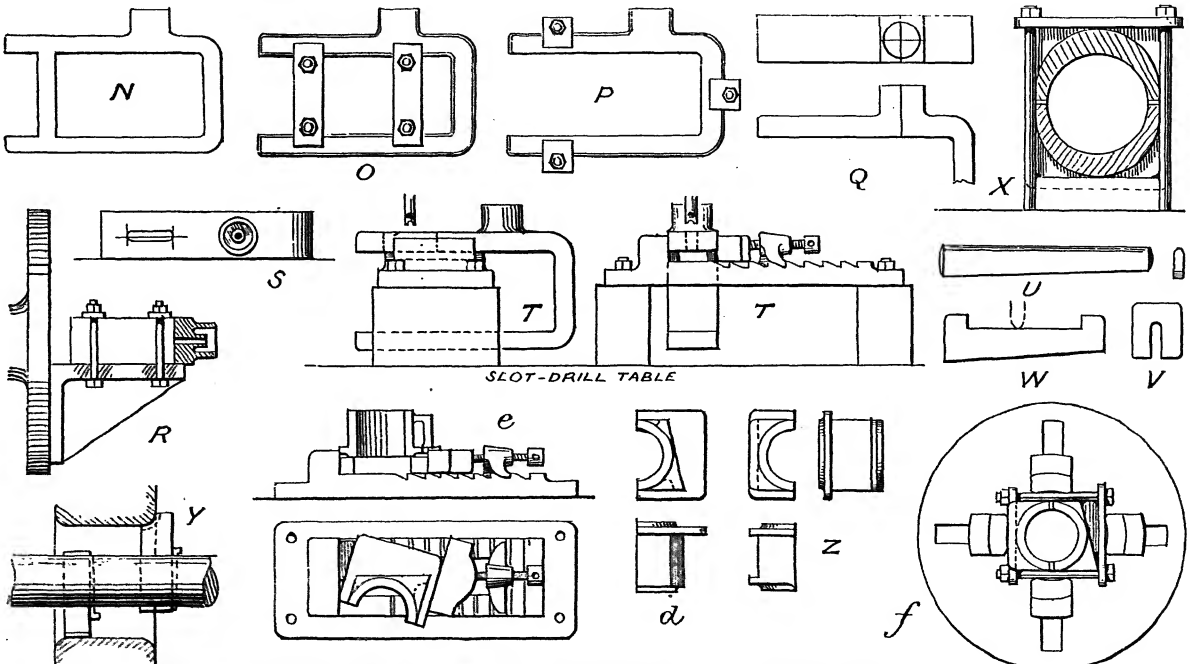
The **Strap** (Fig. 253), being forged fairly to shape, is first scribed to thickness, and planed. A sheet-iron template is next provided of the form shewn at N, Fig. 253, which is placed on the forging and the form traced. Finish the contour with vertical mill or slotting-tool, clamping the work as at O for the outer and as at P for the inner tooling. The oil-cup is next marked off as at Q, and the strap clamped to an angle-plate as at R for turning, boring, and drilling. At the same time the screw is chased for the oil-cup cover. Lastly, line out the cotter hole as at S, and slot drill by blocking up in the machine vice as at T, and, on removal, draw-file and polish.

The **Cotter** U is first planed to thickness from good steel, and then marked off to length and width. Both edges are then planed to the marked lines, and the rest finished very exactly by file, with the aid of the gauge template V, great care being taken regarding the thickness.

The **Gib** W is similarly marked out, and the sloping edge planed. The channel is then removed with a shaping tool, several gibs being bolted together for economy, and the rest finished very carefully with the file.

The **Large Brasses** are marked off and planed in the same way as were those for the bearing, Fig. 249, and are then bolted down very firmly to the boring table as at X, with liners between to represent the draw of the cotter, and with bolts lying close up to the outer surfaces. See their faces are set at right angles to the boring bar, which is inserted as before, and the work traversed into position. Bore right through, and finish the radii with a specially ground tool, as at Y.

The **Small Brasses** are shewn at *d* and Z. They are planed as before, with the exception of the sloping side, which requires a new setting, as shewn at *e*, and is planed with a side tool. The 'ring' faces must also be left untouched, these being turned at the same time as the hole is bored, which is done by bolting the two brasses together, with a wedge between for the

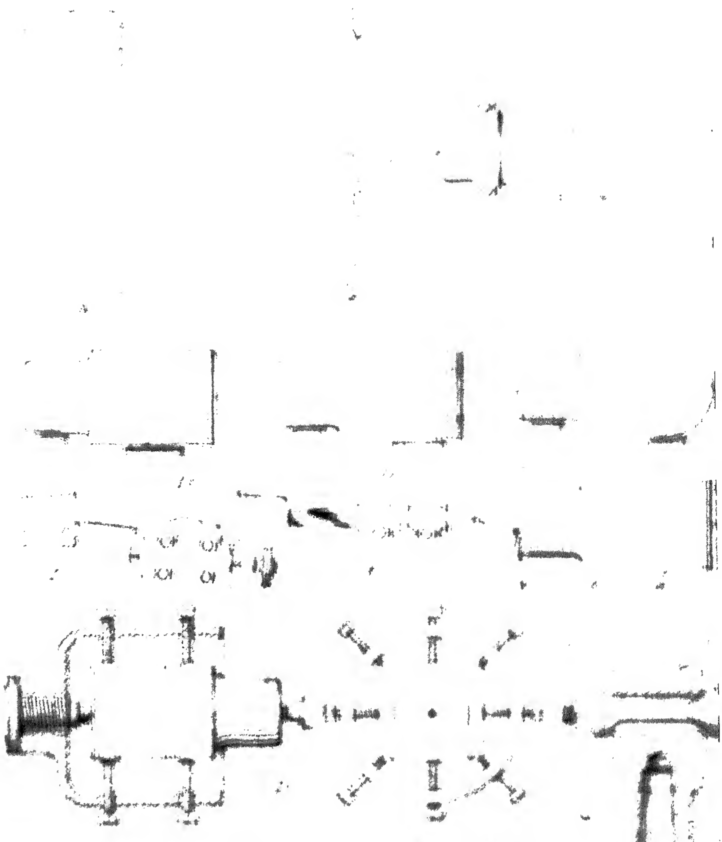


XXI a. Connecting Rod Details. Fig. 253.

slant edge, and a liner to represent adjustment allowance, and the whole chucked in the lathe, as at *f.* Two settings are of course necessary. (*See Appendix I., p.* 752.)

The **Wedge** is now shaped to dimension, but not drilled; the wedge bolt and set screw for the large end both prepared. The oil-cup cover is then turned and screwed in a concentric chuck, and milled on the hexagonal faces with a horizontal tool by placing the work on a dividing plate. All is now ready to put together. For the small end, fit the brasses in place by smooth filing and scraping; fit the wedge, and mark off the hole for bolt by scribing through the rod end. Remove wedge for drilling and tapping, then replace. For the large end, the gib and cotter are first carefully fitted to their holes separately; then the brasses are fitted to the strap, and the latter to the butt end. Place all together, and file the cotter till it enters the proper amount; then mark off the split-pin hole and drill. Once more replace all parts, and the connecting rod is complete.

XXII. Crosshead (Fig. 254).—Centre the forging, as at B, and line the width across the cheeks; then turn the side and end, and shape the flats. Lay now upon the marking table, as at C (see both views), and scribe the horizontal centre line. Find the centre for the gudgeon hole, as at *a* and *b*, measuring from a straight edge, and test also with dividers; erect this line with square, and strike the circle on both sides, also the contour of the boss. Chuck in the lathe, as at L, and bore the hole, first drilling to admit the boring tool. Remove from the dogs, and insert next in a large bell-chuck, as at D, the exact position being found by placing the work between the lathe centres; afterwards firmly tightening up the screws, as shewn. First drill the hole as large as allowable, the tool being centred, as at F, and clamped in the slide rest; and next bore the taper, as at E, by turning the top slide of the rest to the required angle, the feed being obtained by a small pulley on the screw, driven from the countershaft by catgut band. The hole is tested for diameter with callipers, and the angle of the rest noticed before removing (this being afterwards required for the Piston Rod). Now place the crosshead on the mandrel of a shaping machine, as at G, and shape all round up to the return curve, the latter being tooled with a



[illegible]

XXII [illegible]

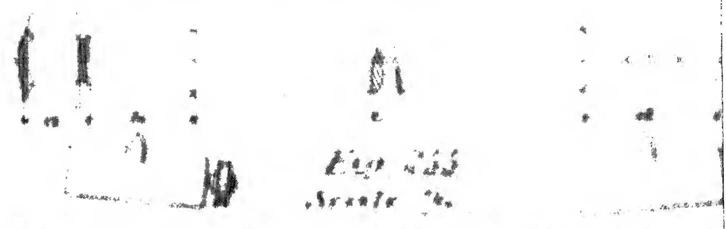


Fig. 553

Scale [illegible]

XXIII Piston Rod [illegible] Forged Steel

Machined all over

concave feed (mentioned in Chap. V.), and the flat portion with horizontal feed. Take again to marking table, block level, as at H, and scribe both fork and slot-hole, measuring from shoulder. Drill and slot the fork, and slot-drill the cotter hole. Finally, slot out the key way to suit the gudgeon; prepare a cotter, and take out the taper in the hole with round file; then draw-file and polish. (*See Appendix I., p.* 752.)

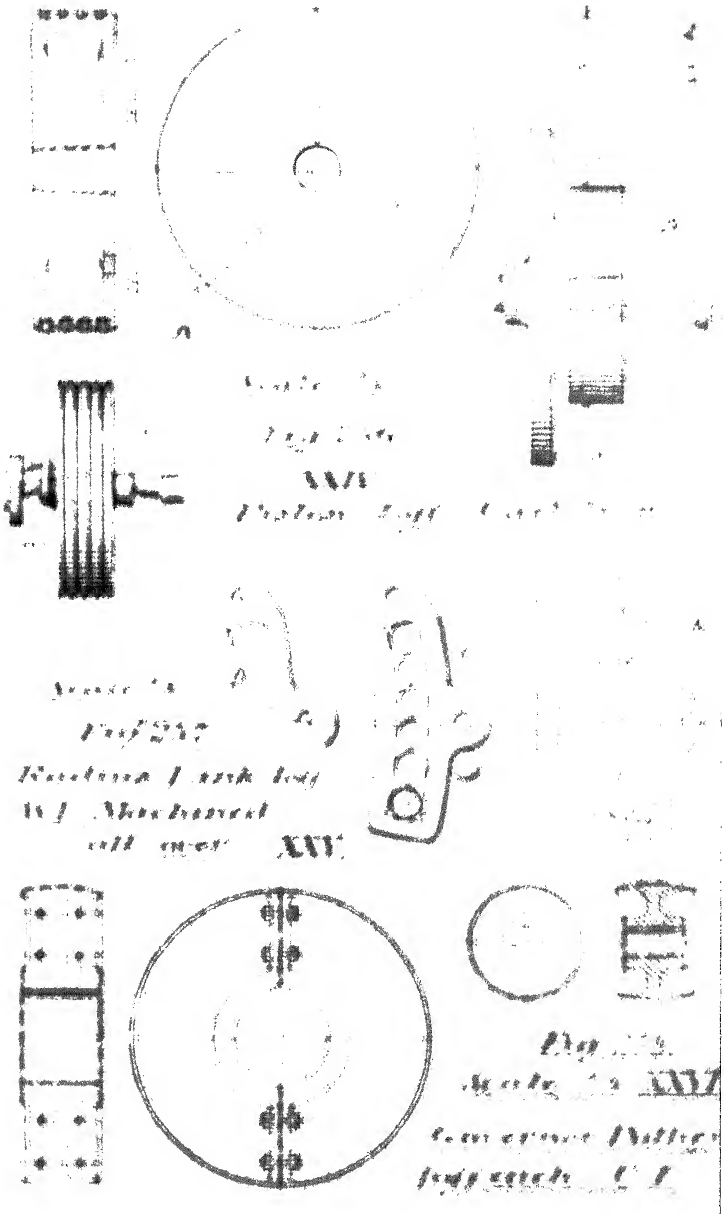
XXIII. Piston Rod (Fig. 255).—Centre on **V** blocks, and set in the lathe. Then traverse all over the work to the diameter at *c*. Mark off the various lengths *a*, *b*, *c*, *d*, and put a centre pop at each place. Turn down *d* to the larger diameter, and take down the taper at *b* and *d* by setting the slide rest, as at E (Fig. 254), and it should be noticed that if the rest be placed at the same angle both for rod and hole, the one is bound to accurately fit within the other. Turn down at *a* to screwing size, and chase; then finish and polish the whole.

The *Nut* may be turned, bored, and screwed in a chuck, and the hexagon milled. Lastly, the rod is fitted into the crosshead, and the cotter hole marked through to the latter, then slot-drilled, and finished with file.

XXIV. Piston (Fig. 256).—This is to be turned on the rim, and bored to fit the piston rod. The latter operation is done at B, and the former upon a taper mandrel at C, the grooves being turned at the same time to exact gauge, so as to fit the rings as truly as is consistent with freedom. The plug holes, *b*, left during casting (see B, p. 30) are to be drilled and tapped, centres unimportant, the plugs being made from a round bar, screwed in the lathe and parted off to length. They may be an easy fit in the holes, but must be painted with sal-ammoniac, so as to form a rust joint.

The rings are rolled from $\frac{1}{4}$-inch brass bar, being received at the works ready formed, sprung out to a somewhat larger diameter than the cylinder. The joint is shewn at *a* (Fig. 256), and should be as nearly as possible closed when the piston is in place.

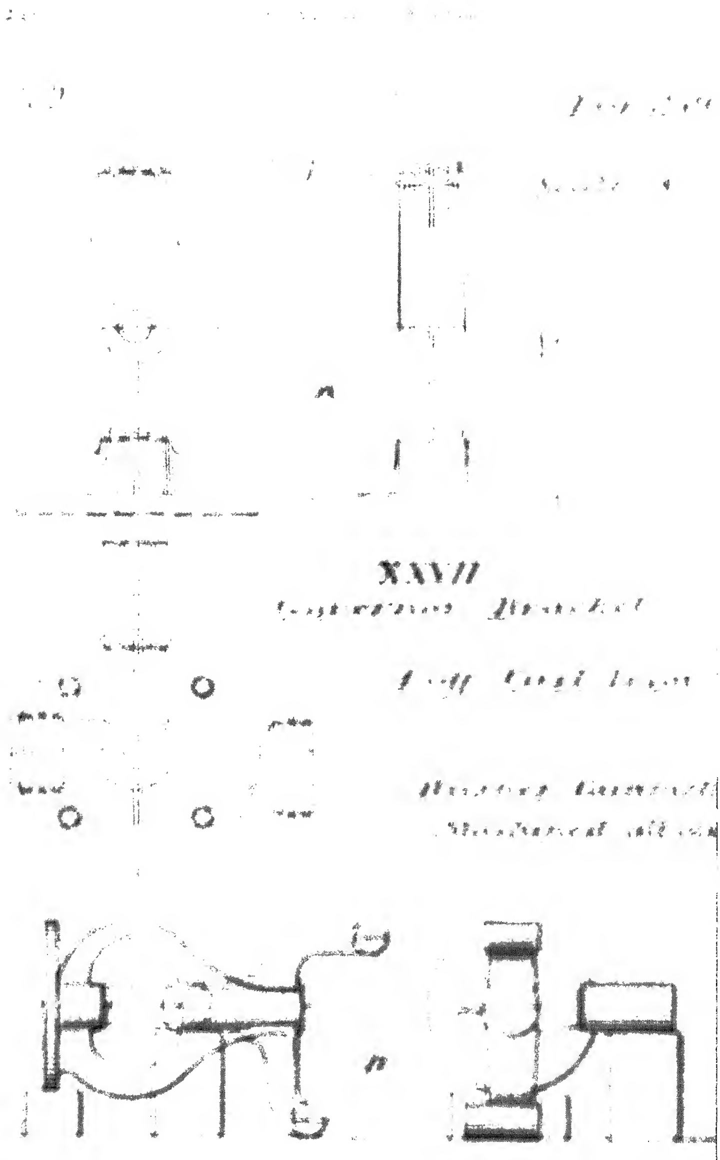
XXV. Radius Link (Fig. 257).—The forging should be fairly to shape, being made to template. First line to thickness, and plane. Make a template exactly to drawing A, with the exception of the holes, which consist of quarter circles, as at B, with


XXVI
Machined
all over XXVI
Scale XXVI

a little piece filed out at the centre to admit the scriber. Lay this template on the work, and trace out. Then drill the holes, which are to be broached when all the parts are put together. Remove all the outside material with a vertical milling tool having a radius equal to that of the return curves, as shewn at C. The inner slot may be cut out by one of three methods: (1) Let several holes be drilled, as at C, one large enough to take a slotting tool, and slot all round with hand feed; (2) Drill a hole to take a vertical milling tool, shewn at C, and a few more holes to save the cutter, and mill out the rest, traversing by hand, first one side and then the other; (3) best of all, is the same as the last, with this exception: the cutter is held in a special form of vertical mill, called a 'profiling' machine. Here the bearing carrying the vertical spindle may be made to traverse any particular curve by applying to it a copy of the same shape, and its action is thus similar to that of the copying lathe. The curve would thus be finished right off without further filing; and the ends may be taken out with a double corner tool, then finished by hand. The die (Fig. 242) is ultimately fitted to the link by careful filing and scraping, and both link and die (after broaching the former) are case-hardened. (*See Appendices I. and II., pp.* 752 *and* 819.)

XXVI. Governor Pullies (Fig. 258).—These are to be machined as shewn upon the drawings. The bosses are to be bored by chucking in a dog-chuck, and the facing both of boss and rim done at the same time. Two settings are, of course, necessary. Next put the small pulley on a plain mandrel, and the large pulley on an expanding mandrel, as shewn at page 155, and turn the rim surface in each case with parallel traverse: then finish the curve with a hand tool. The large pulley being in halves must first be planed on its joint surface, as in the case of the eccentric straps at Fig. 245, then drilled for the bolt-holes, and bolted together for boring. The keys are lastly taken out by slotting.

XXVII. Governor Bracket (Figs. 259–60).—This is a rather more difficult example of lining out, but involves no new principle, the only precaution of importance being very careful levelling at every operation. The casting is laid on its side, as in the two views at B, and adjusted until the bush centres are of the



same height. The three bosses are next faced on each of the ends until as nearly level as possible. No doubt the casting will

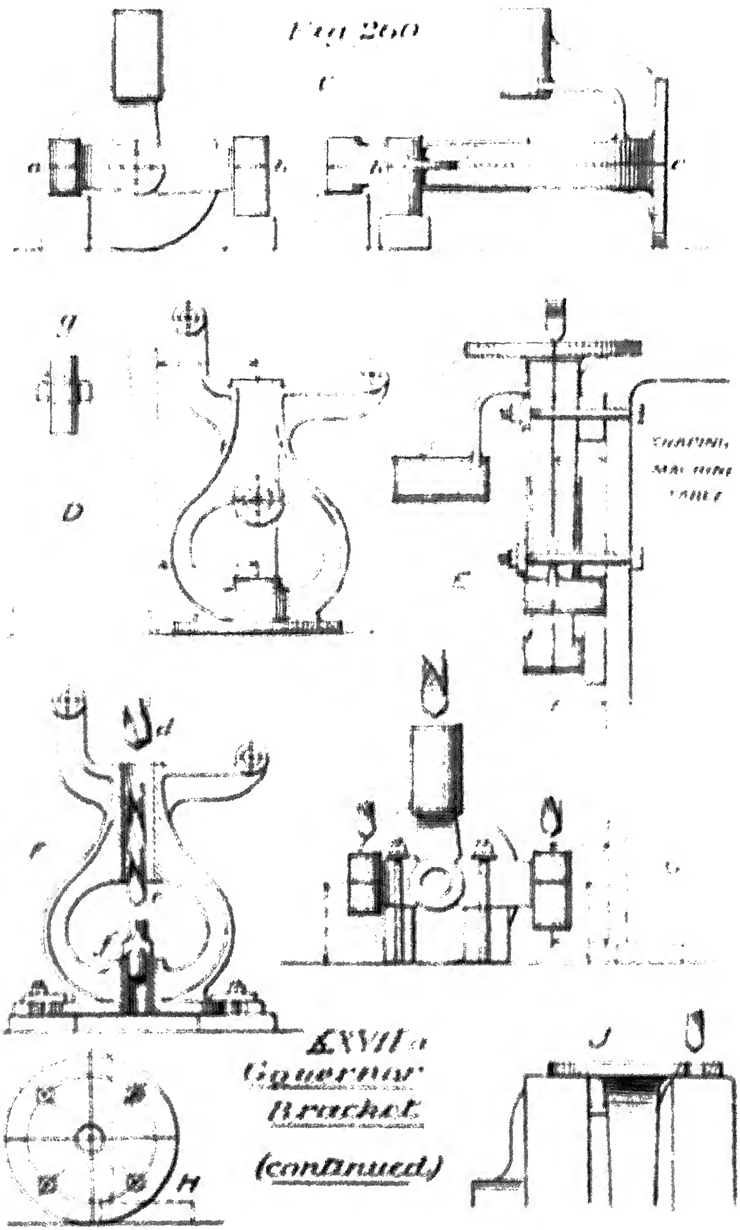


Fig 260

XVII a
Governor
Bracket
(continued)

[illegible]

surfaces with the tool shown at g. The bracket is now clamped to a radial drill bed, as at [illegible], setting by means of the centre line permanently scribed, the holes are drilled to finishing size, and the top surfaces faced. The work must then be reversed to face the bottom surfaces, but not necessarily clamped this time, for the holes themselves will be sufficient guide. The bolt holes may next be marked as at m by laying the bush-holes, scribing and squaring. The bolt circle is then marked, each angle bisected, and the holes drilled at p, supporting the work on blocks. The top bush for the pulley spindle is simply drilled, and after being turned on a mandrel, then driven into its home with a hand hammer. The lower bush is also placed on a mandrel for turning, but having a blind hole must be centred on **V** blocks.

XXVIII. Governor Details. The various parts are shown at Fig. 161, and will be taken *seriatim*.

The **Spindle** is first centred, turned, and screwed, and the

flat cheeks lined out. These being shaped, the boss is next marked off and milled, and the hole drilled, the slot taken out as in previous cases, finishing by hand. The key for the mitre wheel is finally grooved with a milling cutter. The **Sleeve** B, being cast solid, is first centred, and the thickness of the bosses lined. It is drilled in the lathe, and then slipped over a mandrel to turn, and to screw the end. The flat surfaces of the bosses are next shaped across, and the space between taken out with a tool of the exact width. The holes are marked off and drilled, and the bosses filed round, after which the sleeve is fitted to the arms C by chipping out the socket with cross-cut chisel and finishing with a curved file much used by brass finishers, called a 'riffler' (see Q, Fig. 262). Mark out key-way for the weight E, and cut the same by hand.

The **Nut** for the sleeve is bored and screwed in a chuck, and turned on a mandrel, and the octagon milled by fixing on a dividing circle. Drill and tap for the side screw, but only file out the corresponding slot in the sleeve after M is put into place, and the nut advanced to give the requisite tightness. The **Lower Arm** C is packed up as at H, Fig. 262, and the fork bridged; then the centre and the flat cheeks are lined, the fork centres struck, the lengths marked off, and the centres of the bosses squared up. Next turn the shank, and slot or mill the fork to the marked lines. Lay the arm in the position J, and after scribing the centre line, strike the curves of the bosses and pin-holes, and scribe the width of the fork. Shape and mill to the lines, and drill the holes. The **Radius Arm** D is centred and lined as at K, Fig. 262, and the shoulder line marked off, as well as the commencement of the small curve to ball. Set in the lathe, and turn down the shank. Then prepare a template for the ball, as shewn at L, Fig. 262. First turn to diameter as a cylinder, and surface the end to length; then feed at 45°, as at *g*, Fig. 262; continue to halve these angular feeds until the ball is approximately spherical, as tried with template, and finish with a keen hand tool ground to the ball curvature. Mark the centre of the ball while revolving in the lathe, and set on marking-table to get the cross centre, as at M, Fig. 262. The boss is then finished as usual, and the hole drilled through the ball. The

Machined all over.

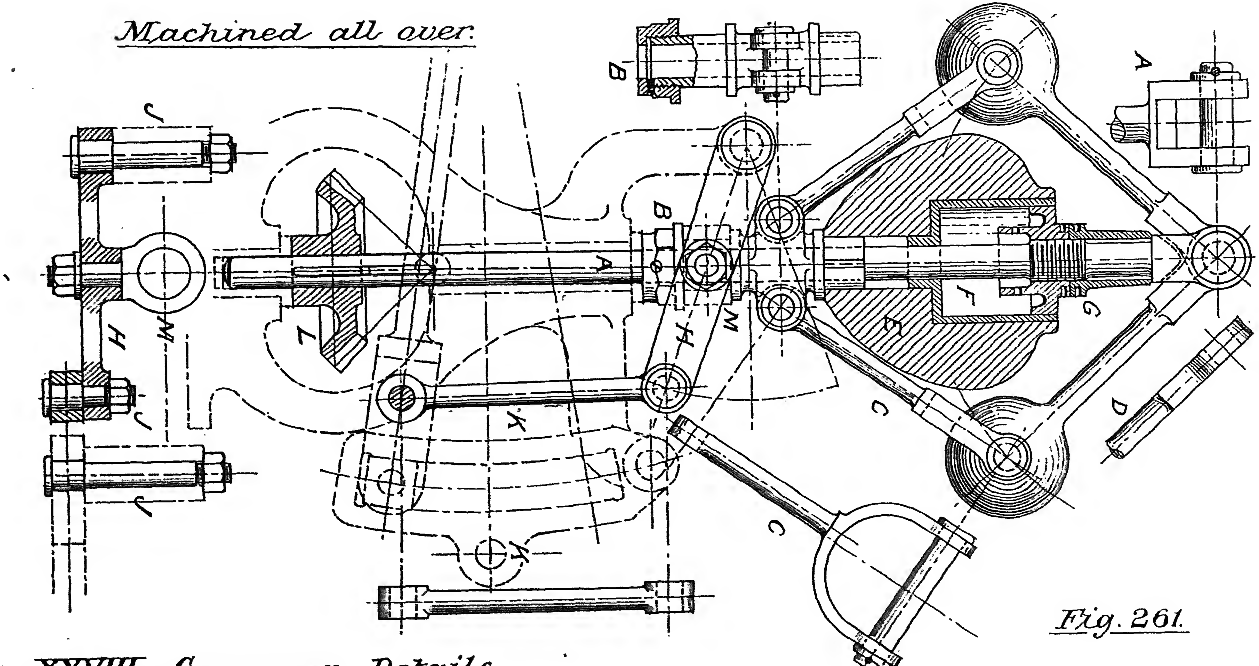
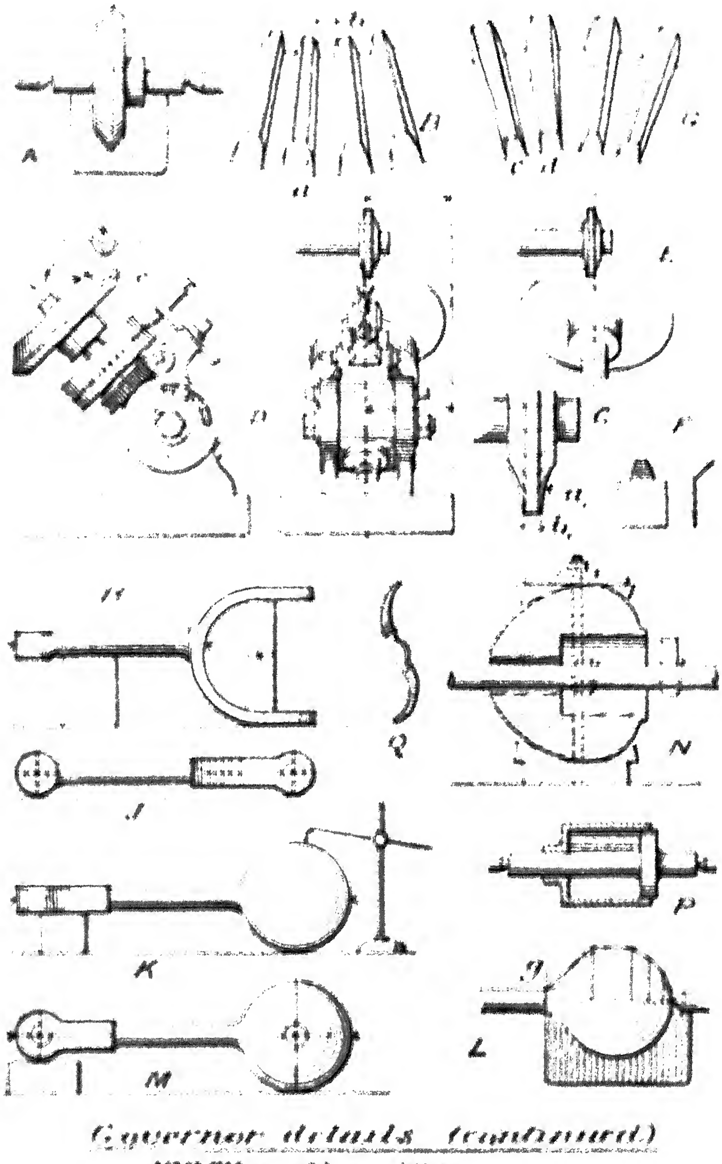


Fig. 261.

Scale 1/6.

XXVIII. Governor Details

1 set off. complete.



Governor details (continued)

XXVIII a Fig. 262

Central Weight E is fastened to the table of a horizontal boring machine, as shewn at N, Fig. 262, and bored with cutters of correct radius. It is next put on a mandrel fitting the smaller hole, and the outside turned to template. First the ends are faced, then the diameter turned as a cylinder, and the rest is obtained by various angular feeds, finishing by hand. The key-way for fastening to sleeve B is to be slotted. The **Bush** F for the weight, is to be bored in a chuck, and then turned on a special mandrel, shewn at P, Fig. 262, being afterwards driven tightly into the weight by means of a copper hammer. The **Nuts** and **Guard** G are first bored, and afterwards turned on a mandrel, being replaced in the chuck for screwing. The tooling of the **Lever** H may be understood by reference to the regulator lever No. I., and the studs J J are all examples of simple turning and screwing. The **Lifting Link** K, and the **Lifting Eye** M, need no special description. (*See App. I. and II., pp.* 753 *and* 820.)

We now come to the **Mitre Wheels** L. For the machining of these we may again refer to Fig. 262. Both wheels are made of gun-metal and are exactly alike, boss included. After boring truly they are placed on a mandrel, and the 'blank' turned as at A to a template which has been previously made with great care. The teeth are then to be cut by means of a milling cutter. A mandrel is provided which fits into the socket of the dividing centre shewn at D, and the wheel set at such an angle that the *lower* line of the tooth, *e f*, is horizontal. Looking in front of the wheel, the work must be set so that one *edge* of the milling cutter is in line with the centre. The radius a_1 of the cutter at C being made to fit the curve a of the larger end of the tooth, as shewn at B, and the width b_1 of the bottom of the cutter made equal to b at the narrow end of the teeth, a little consideration will shew that the cutter will trim up one side of the tooth in such a way that the smaller ends of the teeth d will be a little too wide at this point, as shewn at G. After all the spaces have been cut out as at D and *one* side of the tooth, the work is traversed forward and the other face cut as at E, after which the taper C of the teeth is lined out as at G, using (1) a straight edge of the form shewn at F, page 62; and (2) a template F, Fig. 262. These surfaces are then dressed off with the file. (*See App. I., II., V., pp.* 753, 823, 986.)

XXIX. Steam Cylinder (Fig. 263).—The various operations are shewn at Fig. 264. The ends are first bridged, and the centre found by reference to the outer curve of the cylinder flange. Mark temporarily the height of the centre B. Adjust until the top of the cylinder foot is fairly level, giving and taking with the three centres at A and B. Scribe the horizontal centre line, B B, all round, and square up the vertical line, C C; then strike the circle D for boring. Line the heights of the steam and exhaust flanges at E, and scribe the thickness of the foot at F; line also the thickness G of the bosses for the bolts. Scribe the height of the valve-guide H, using a special piece of bent wire for the scriber as shewn, and mark the heights, J, of the indicator bosses.

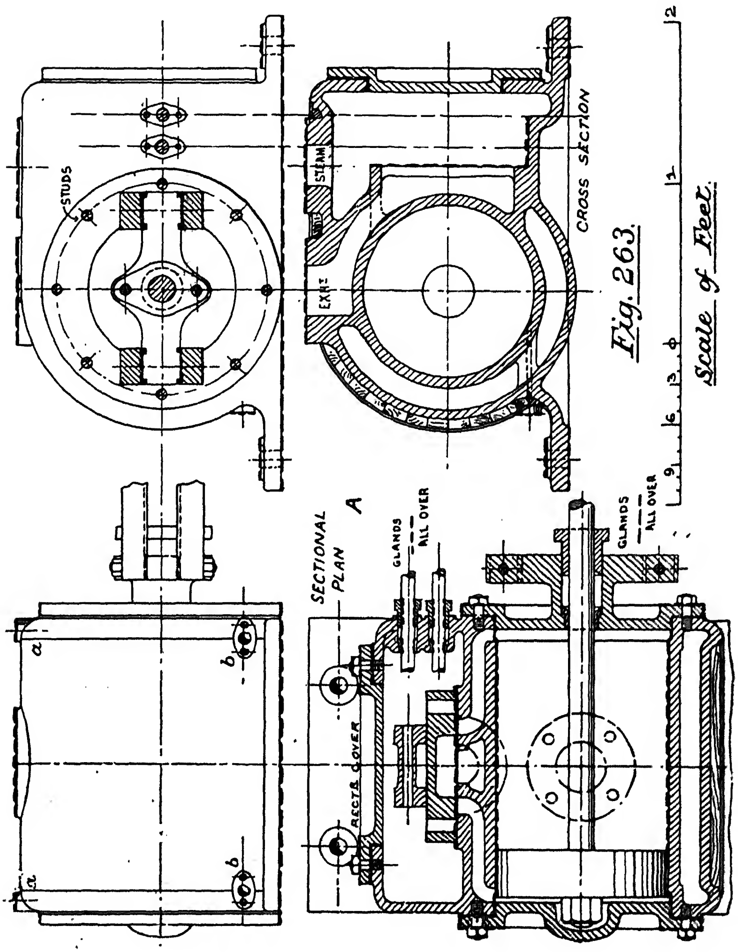
Set the cylinder upside down, as at M, and plane the foot. Set upright, as at N, and plane the steam and exhaust flanges, the indicator bosses, and the foot bosses. Now clamp between angle plates on the planing machine as at P, and if these be true vertically (as they should be) there will be no difficulty in packing correctly; but if not, some care must be exercised, and in any case the centre of the cylinder must be levelled longitudinally. Scribe the steam chest face to correct distance from cylinder centre, and plane with a front tool, P. At the same setting the valve face may be planed at Q with long, strong side tools, right and left, and the valve-guide also finished.

The cylinder must next be bored. This is done by packing up, as at K, on a horizontal boring machine of the type described on page 161, but of a smaller pattern. Bore right through, and face the flanges by first measuring through the cylinder, as at L, so as to leave an equal amount of seating on each flange; then alter the tool to take out the bell mouth or larger diameter at each end. Set the casting on end, as at O, and plane the stuffing-box seatings so as to be level with the cover seating.

All the main faces are now machined, and the rest of the lining may be done. At V the inner square is scribed on the steam chest face, measuring from the horizontal and vertical centres, the latter being squared up with reference to the outer edge of the flange. Take the material out by hand, or by slotting tool, very probably the former. The steam chest cover may be

XXIX

Cylinder for Horizontal Engine. 1off. C.I.



planed in the same manner as the cover at Fig. 237, and the holes drilled. It may then be fitted to its place on the cylinder, and the holes scribed through, as at v, and centred. After the front and back cylinder covers have been finished (next example),

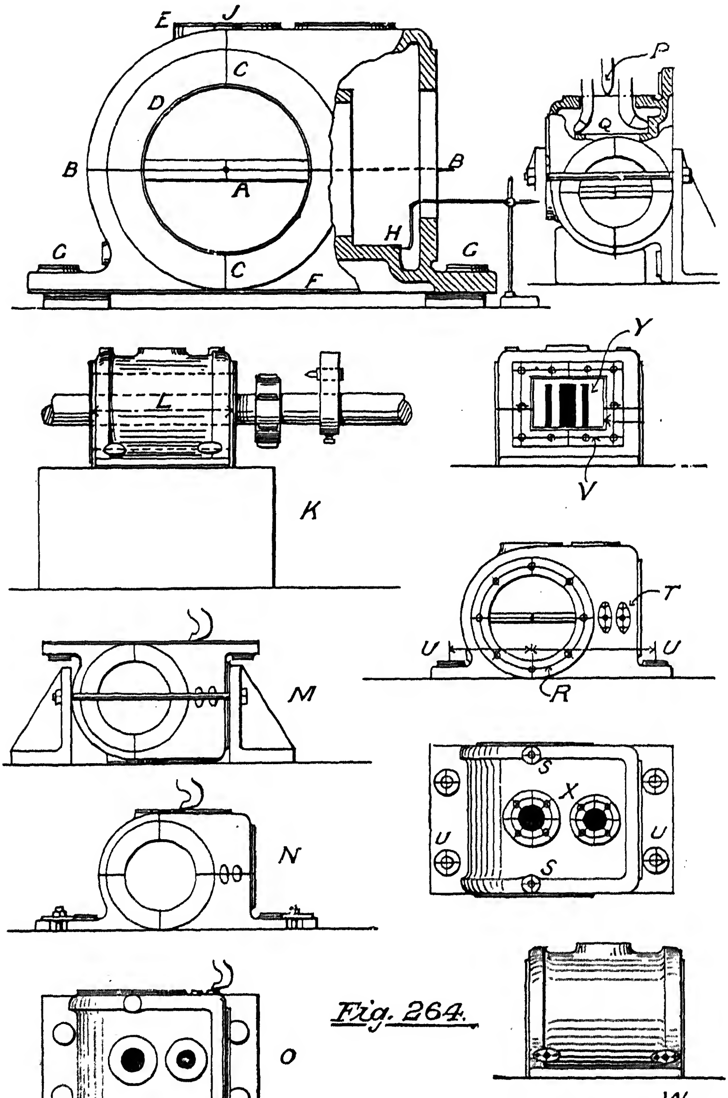


Fig. 264.

Steam Cylinder

(continued) XXIX a.

[illegible]

[illegible] face after facing, the template is then squared, and the [illegible] traced through.

Now set the cylinder [illegible] stud holes on the various flanges to tapping size, and tap them either by hand or by machine.

The indicator bosses have a small hole drilled first, are [illegible] afterwards, [illegible] down to tapping size, and the [illegible] their top surfaces, then finally tapped. The [illegible] are treated in like manner, and the cylinder foot bolt holes [illegible] on their top surfaces after drilling. Next [illegible] the [illegible] steam cocks and valve spindle stuffing boxes, the [illegible] (fig. 237). Chip the edges of the steam ports to the scribed lines and scrape the face. Then put in all the cover studs with [illegible] lead, and bolt on the covers to try the fit.

XXX. Cylinder Covers (fig. 243). The front cover is chucked, as at a, until the stuffing box face be true, but that is not to be finished. Set the tool to correct diameter from centre and turn the rim. Clean up also the front of the cover, [illegible] from the chuck [illegible], and mark off the thickness of flange, and the circle for the stud holes. If cored, [illegible] bore; if not, drill first, and bore afterwards to diameters given, first right through with the smaller hole, and next the larger

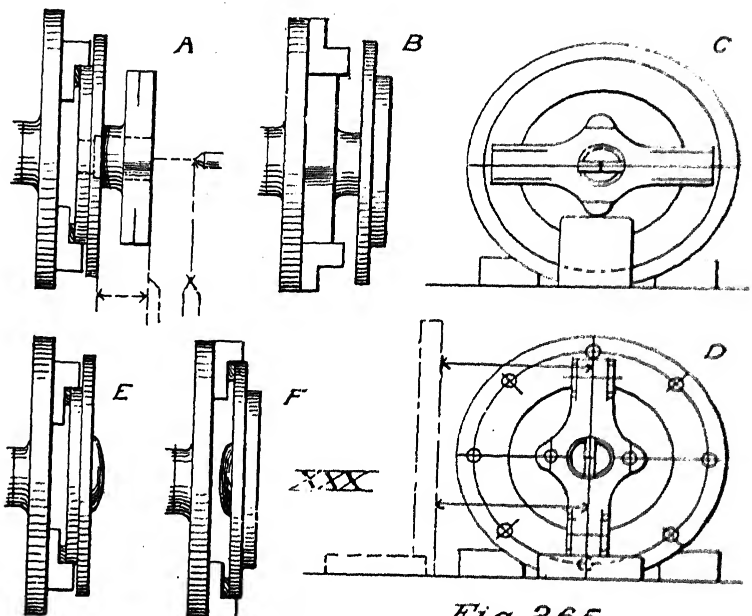


Fig 265.

Cylinder Covers (continued)

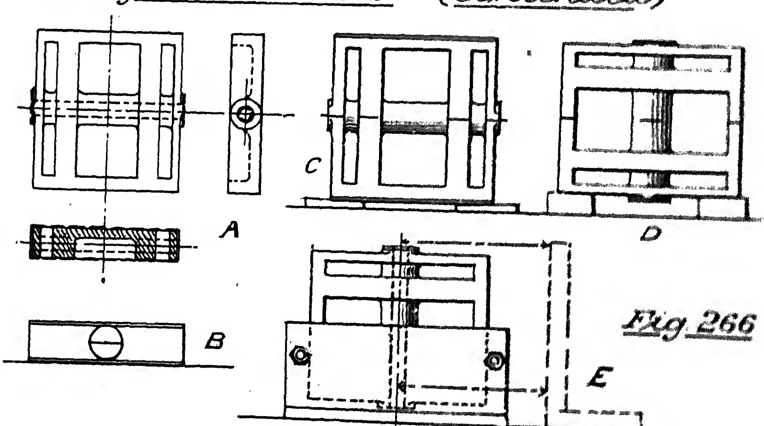


Fig 266

XXXI. Main Slide Value.

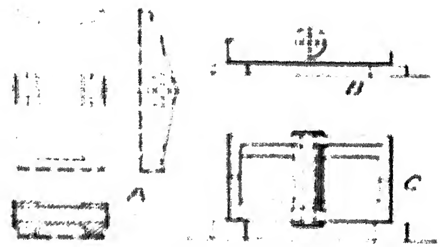
1 off Cast Iron

to proper depth. Reverse the work, as at B, and set truly with the turned face. Turn the inner surface and the smaller diameter. Now set the cover, end up, on the marking table, as at C, until the boss is horizontal. Having spanned the hole and found its centre, scribe the centre line along the boss and on to the cover. Scribe also the seating for the slide bars and the centres of the gland studs. Now turn the cover through 90°, by measuring from a square, as at D, and scribe the centre line across. Mark also the slide bar seatings, and divide out the stud holes; then drill the bolts and studs, and slot the slide bar seatings. Lastly, scribe through the stud holes on to the cylinder flanges. The back cover is turned, as at E and F, and similarly marked, and the gland tooled as previously described at Fig. 238.

XXXI. Main Slide Valve (Fig. 266).—Lay horizontal, as at B, scribe centre of boss and thickness of valve, and plane both sides. Set up, as at C, till the bosses are level, and scribe the centre all round; line also the top and bottom surfaces. Turn to the position D till the boss is quite vertical, and scribe the centre line round. From this, line the height of boss at top and bottom. Re-scribe the hole for the spindle at both ends, which is much larger than the valve spindle, to allow for wear of valve. Next set up on an angle plate in the drilling machine, as at E, till vertical, and drill the hole right through, knifing at the same time. Plane the top and bottom surfaces. Line out the ports by means of a template, and finish their edges by hand.

XXXII. Expansion Slide Valve (Fig. 267).—Set level, as at B, and scribe the boss centres and the face. Square up the edge, measuring from the centre, and join this along the top. Next set vertically, as at C, scribe the centre, and line thickness of boss seatings. Drill, knife, and plane as before, finishing the edges to template.

XXXIII. Flywheel (Fig. 268) requires very little description. It is simply bolted centrally by the arms to a large face plate, as shewn at A, the boss bored, and both boss and rim faced. It is next reversed, the other side faced, and the rim turned, as in previous similar cases, the curved surface being given by a careful hand feed. (For keyway cutting, *see Appendix I., p.* 754.)

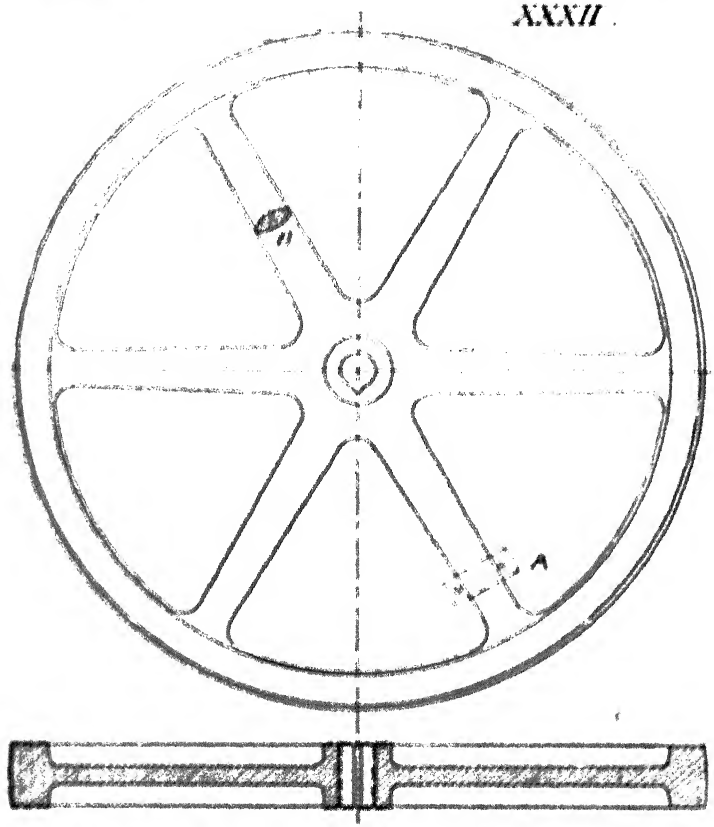


Expansion Slide Valve

Loff Cast Iron

Fig 267

XXXII.



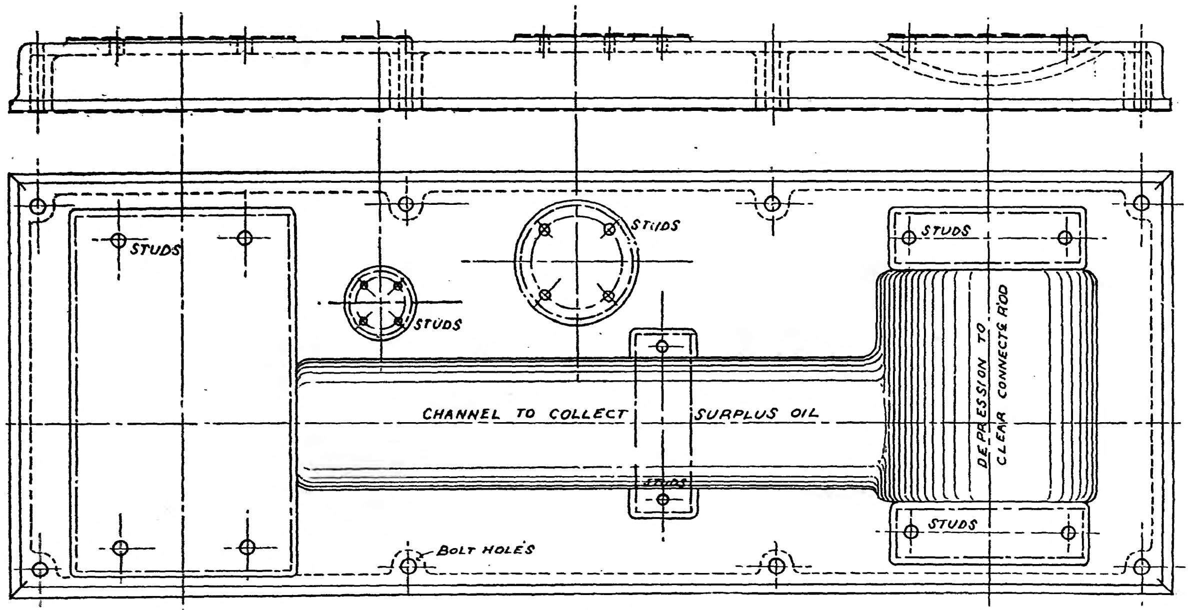
XXXIII *Fly Wheel.* *Loff.* *Cast Iron.*

Scale 1/24 *Fig 268.*

XXXIV. Bed Plate (Fig. 269).—This is not too large for a planing machine such as was described at page 169. It is better to plane the under edge, so that the bed may rest more perfectly on the stone or brickwork. The casting is therefore set upside down on the machine, and the ends clamped till the two side edges are planed; the clamping is then removed to the side, and the end edges planed with a short stroke. The bed being now set right way up, and held by its lower rim all round, must next have its seating marked, so as to plane off the calculated allowance (the total depth is not of any consequence). All the seatings will be done at once, with a stroke the whole length of the table. The bolt and stud holes are to be marked off by the Erector.

XXXV. Brass Work (Fig. 270) must be bright all over the exterior, and have the interior bored at certain after-mentioned places. The **Oilcup** at B can be finished entirely by chuck turning and drilling, polishing with the very finest emery cloth. The **Cylinder Cock**, A, is cored throughout. The main body and the plug socket are both turned externally as far as possible, but the central portion must be finished with file, and the corners cleaned with a riffler. The socket and plug are respectively bored and turned in the manner shewn at Fig. 254, the cock then placed in the vice, and the plug ground to fit, with fine emery powder and water, by rotating backward and forward with a wrench upon the shank. The screws are chased, and the flange drilled; and the whole polished with fine emery cloth. The union nut, after finishing, is slipped over the copper pipe, and the conical nipple then brazed to the latter (see page 86).

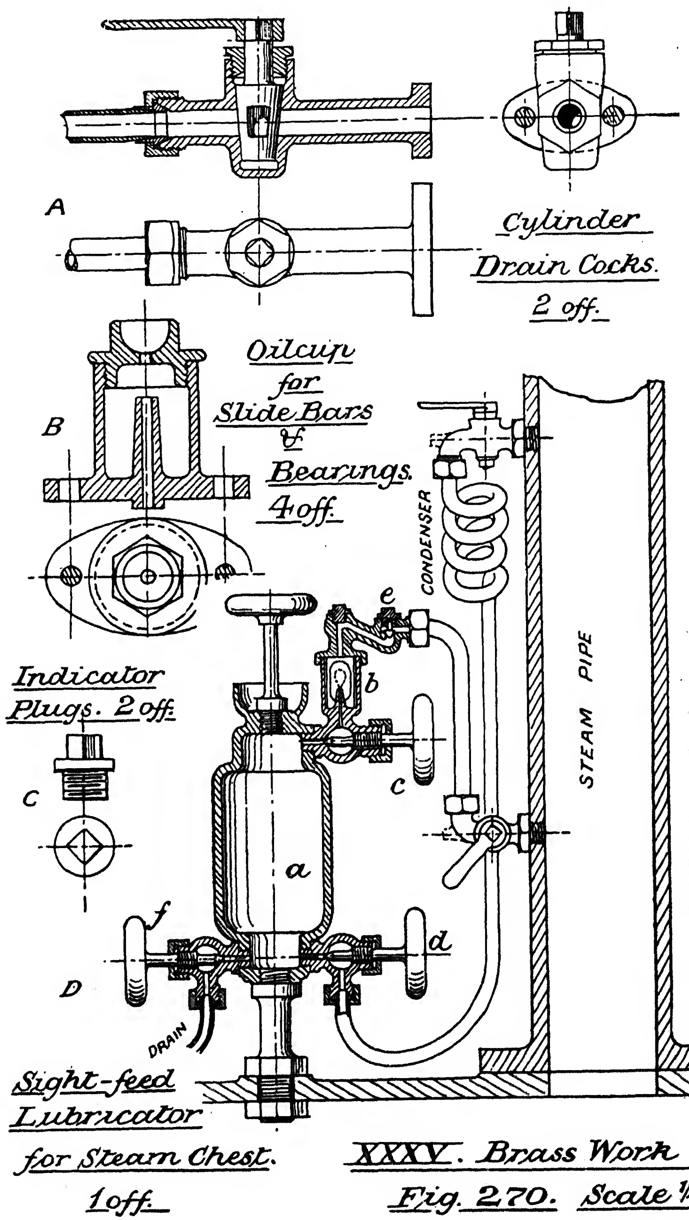
The **Sight-feed Lubricator**, D, is the only form now used for slide valve and piston lubrication. The oil-chamber, *a*, is fixed in any convenient position, and two connections made with the steam pipe as shewn. Having filled *a* with oil, and the sight-feed *b* with water, the valves *c* and *d* are opened, as well as the two steam cocks, and steam being condensed in the coiled pipe, forms water, which enters *a* and displaces the oil, forcing it up through the glass sight-feed chamber drop by drop, it being seen rising through the water in *b*, than which it is specifically lighter. Reaching the steam pipe, it is carried by the steam to the slide



XXXIV. Engine Bed Plate 1 off. Cast Iron. Fig. 269.

0 1 2 3 4 5 6

SCALE OF FEET



XXXV. Brass Work
Fig. 270. Scale 1/4

valve and cylinder; *e* is a non-return valve, and *f* a drain cock. The various parts are bored, screwed, and polished, and then put together. The steam cocks are cast with a core, and are provided in casting with a small boss placed on the bend to assist the centering in the lathe; this boss is shewn dotted.

The **Indicator Plugs,** C, are next turned and screwed.

Erecting.—We may now collect all the engine parts for the purpose of erecting, as follows:—

LIST OF ENGINE DETAILS.

		Figs.
I.	Regulator Lever	228
II.	Bracket for Regulator Lever	229
III.	Pins and Washers for Regulator Lever	230
IV.	Links for Regulator Lever	232
V.	Regulator Valve Spindle	233
VI.	Nut for Valve Spindle	234
VII.	Regulator Valve	235
VIII.	Regulator Valve Box	236–7–8
IX.	Valve Spindles: Main and Expansion	239
X.	Expansion Eccentric Rod	240
XI.	Main Eccentric Rod	241
XII.	Intermediate Valve Rod	242
XIII.	Guide Bracket for Valve Rod ...	243
XIV.	Eccentric Straps and Sheaves ...	244–5
XV.	Slide Bars	246
XVI.	Slide Bar Bracket and Distance Piece	247
XVII.	Crank Shaft Bearing	248–9
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XXI.	Connecting Rod	252–3
XXII.	Cross-head	254
XXIII.	Piston Rod	255
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XXV.	Radius Link	257
XXVI.	Governor Pullies	258
XXVII.	Governor Bracket	259–60

LIST OF ENGINE DETAILS—*Continued.*

The Erector is now to be provided with a 'General Arrangement,' or complete drawing of the engine, in plan and elevation, having certain principal dimensions supplied. This drawing is given in Figs. 271 and 272.

The Bed of the engine is slung, and lifted by travelling crane into position on blocks of wood, as at *a*, Fig. 273, and then levelled with wood wedges and the aid of the square shewn in Fig. 196; the cylinder and bearings then adjusted on their seatings approximately; the back and front end of cylinder bore being bridged with iron bars, the first having a small hole drilled centrally and horizontally, and the second having a central notch in its upper edge (see A and B): a strong, fine string *b* is knotted and passed through the hole, and carried to the front of the bed, where it is pulled tight and wrapped round the support C; the latter being set with one edge agreeing with centre line of cylinder, as measured from the bearing seatings, and having notches, as at D, to hold the string at the correct height. This string constitutes the main centre line, and the front of the cylinder is adjusted to suit by tapping the casting with a hammer, then clamping firmly to avoid accidental movement.

The Bearings are next adjusted. Pass a long straightedge, E, through the brasses, and support it on level blocks till its upper surface nearly touches the string. Clamp the large square F upon E, near the string, and support on block G; then prepare a lath H, to measure the length from cylinder face to edge of bearing brasses, and mark distances on the straightedge E, on each side

of the square, up to the bearing faces (shewn by curved arrows). Adjust bearings till (1) straightedge touches measuring lath; (2) square touches string along its whole length; (3) face of brasses is lineable with measure of straightedge; and (4) straightedge exactly touches bearing brass throughout its length. Then mark the bearing stud holes through upon the bed, and do the same with the cylinder holes.

The Slide Bar Bracket must be placed with reference to slide bar length. It is wisest, therefore, to temporarily fasten the front cylinder cover and the two bottom slide bars. When all are together, as at *d*, the bracket is set to central position by squaring up from its top surface to the string, and the stud holes traced through.

The Valve-Spindle Guide Bracket must also be true with regard to the spindles, so these are put through the stuffing boxes and the bracket, and the latter adjusted by measurement from cylinder face on the one hand, *g*, and from the string on the other hand, using blocked-up laths at *h*. If the spindles do not slide truly, a slight readjustment can be made. Examine also for appearance regarding seating, then scribe the stud holes. The governor bracket comes to the Erector fitted up entirely with governors, links, and pullies. Set up in approximate position, and measure the distances J from the boss faces to the string, these being the most important; adjust to these, and also to distances from cylinder face (*e*) and crank bearing (K). Then test, by measurements at K and G, for parallelism of pulley spindle, and mark off the holes.

The holding-down bolts are lastly marked, all the parts removed, the circles centre-punched, and the Bed Plate either taken to a radial drill, or drilled by ratchet brace, the former being preferable. Tap all the stud holes and insert studs; then return the bed to its erecting position, which need not this time be level, the main adjustments having been made. And now the various pieces are to be put in place in the order we shall mention. First the Cylinder is bolted down, and the Front Cover put on, the Piston inserted, and the Rod passed through; then the Slide Bar Bracket, and the *Bottom* Slide Bars. The Bearings come next, and when fixed have the Crank Shaft laid upon them, with

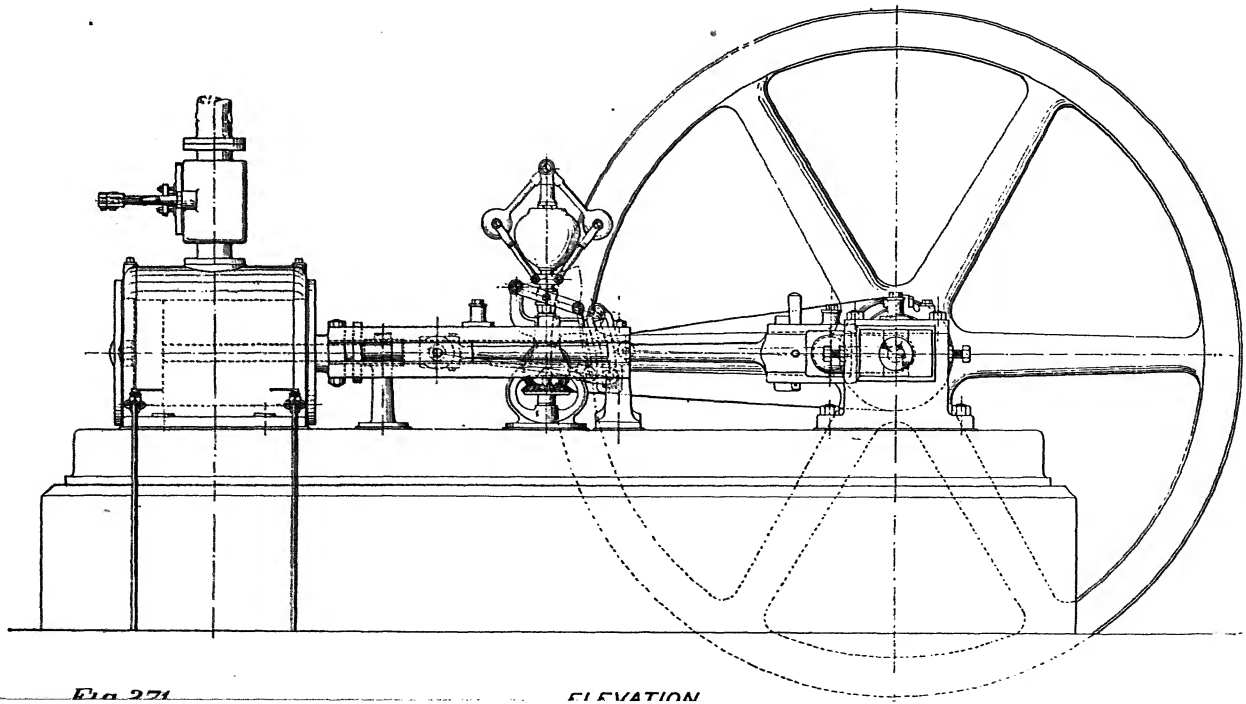
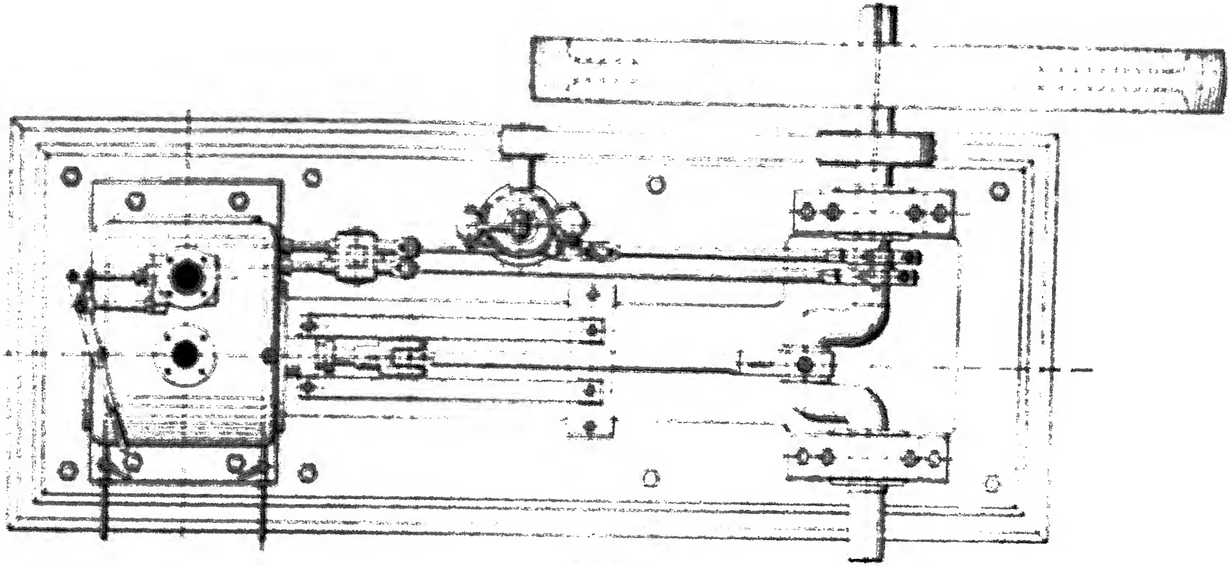


Fig. 271 ELEVATION



PLAN

Arrangement of Non-condensing Automatic-expans. Horizontal Engine

Fig 272

red ochre applied to its journals. Being turned round in the bearings by means of the temporary handles, L, it is lifted away, the brasses scraped, and the method repeated until a perfect fit is obtained. The last time the shaft is removed, it is taken to the marking table to line the eccentric keyways. The angles for each sheave are shewn at M, being known after design, and are there called α and β; x and y are therefore found from a table of natural sines.

For x = radius of shaft × sin α, and y = radius of shaft × sin β.

TABLE OF NATURAL SINES FOR ANGLES UP TO 45°.

Deg.	Sine.	Deg.	Sine.	Deg.	Sine.	Deg.	Sine
½	·00872	12	·20791	23½	·39875	35	·57357
	·01745	12½	·21644	24	·40673	35½	·58070
1½	·02617	13	·22951	24½	·41469	36	·58778
1	·03489	13½	·23344	25	·42262	36½	·59482
2½	·04362	14	·24192	25½	·43051	37	·60181
2	·05233	14½	·25038	26	·43837	37½	·60876
3½	·06104	15	·25882	26½	·44619	38	·61566
3	·06975	15½	·26724	27	·45399	38½	·62251
4½	·07846	16	·27563	27½	·46175	39	·62932
4	·08715	16½	·28401	28	·46947	39½	·63607
5½	·09584	17	·29237	28½	·47715	40	·64278
5	·10453	17½	·30070	29	·48481	40½	·64944
6½	·11320	18	·30901	29½	·49242	41	·65606
6	·12187	18½	·31730	30	·50000	41½	·66262
7½	·13052	19	·32556	30½	·50753	42	·66913
8	·13917	19½	·33380	31	·51503	42½	·67559
8½	·14781	20	·34202	31½	·52249	43	·68200
9	·15643	20½	·35020	32	·52992	43½	·68835
9½	·16504	21	·35836	32½	·53730	44	·69465
10	·17364	21½	·36650	33	·54464	44½	·70091
10½	·18223	22	·37460	33½	·55193	45	·70710
11	·19081	22½	·38268	34	·55919		
11½	·19936	23	·39073	34½	·56640		

The heights x and y above or below centre line have to be scribed by laying the crank webs vertically or horizontally as at P and N, and the distance Q also measured, giving the centre line between the two eccentric rods. Slot-drill the keyways. Then drive the sheaves upon the shaft, to which they should fit tightly; put in the keys, and replace the crank in its bearings. (It may be noted

that the copper hammer should always be used in these operations.) Bolt down the bearing caps.

Fix the Valve Spindle Guide, valve spindle Stuffing Boxes, and Valves, also the Governor Bracket with gear complete. Twist the valve spindles round until the valve screw is placed symmetrically with regard to the valve; then measure for equal play either side of the guide bracket. In the case of the expansion spindle, put in the intermediate rod, and let the lifting link be vertical when valve is at half stroke. We shall proceed to set the valves; so to aid us in turning the crank to its various positions, the flywheel is driven on to the shaft, and there keyed. The governor pulley can be put on afterwards, being in halves.

It is convenient to find the position of the main slide valve by the aid of a thin wedge of wood, R, which is tried in the port on the horizontal centre line, and on removal measured. Put the main slide to open to 'lead' at the front of the cylinder, the amount being known; place the crank horizontal, as taken from the seatings, and put the crank pin to the front, as at S. Now measure with a lath the length from valve spindle pin to nearest edge of eccentric sheave.

Set the valve for lead to back of cylinder, place the crank in horizontal backward position T, and measure the length as before. The two lengths obtained should differ only by a very small amount, and, being averaged, the length of the main eccentric rod can be found. During the preceding operations the expansion valve can be slid to one side or the other for convenience.

The expansion slide must be set centrally. We first move the main slide to opening position at front and back part alternately, and each time measure the distances U on the valve spindle. By setting the spindle to half the sum of the two measurements at U, the valve will be central. The expansion valve is now moved till midway between the main valve ports (V V), and its spindle measured as at U. Set the crank webs upright, as at W, with straightedge and plumblines. Take the distance, Z, to eccentric centre, found thus:

$$Z = \text{radius of eccentric circle} \times \sin \beta\,;$$

and move the expansion spindle back at U by this amount; then

measure length for eccentric rod between pin of radius link and edge of sheave. Reversing the crank, as at X, the valve is moved to the front by the same amount, Z, the length again obtained, and the two averaged. In our description of the machining of these rods we supposed the length already given; but it is always found for the smith in this way, though often the rods are finished in two pieces, and afterwards welded to correct length.

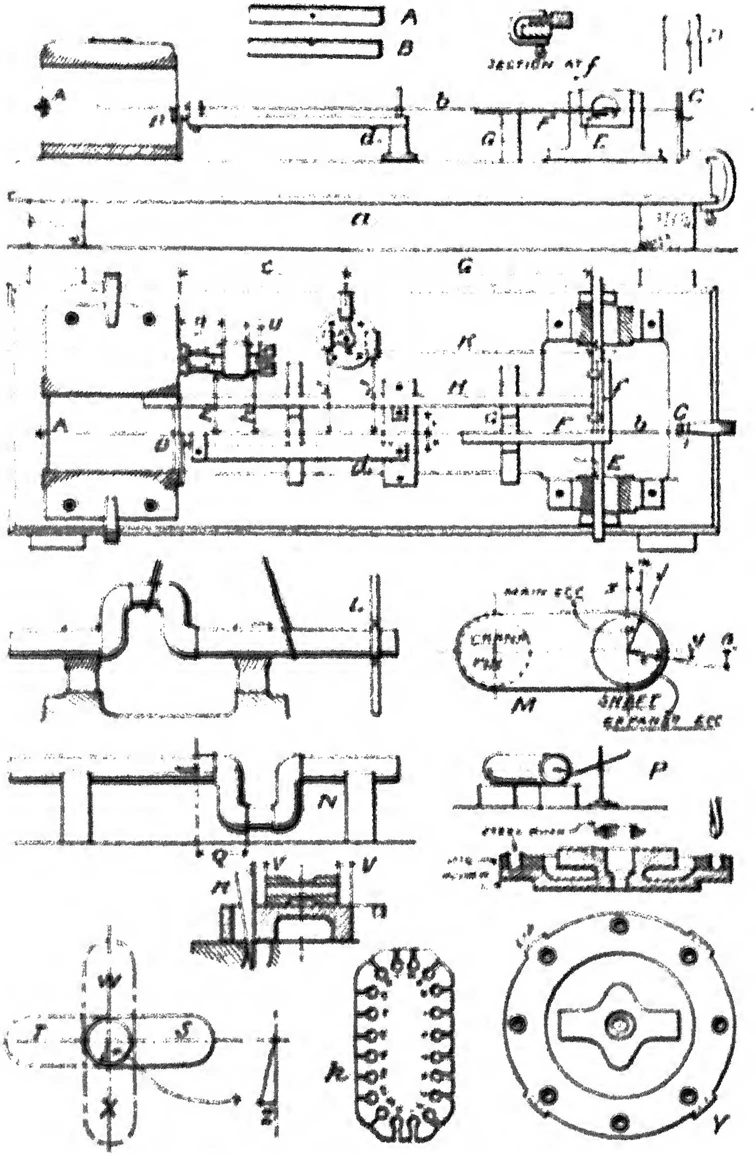
Put the valve rods in place, also the crosshead, connecting rod, gudgeon, and slide blocks; connect up to crank pin, having previously fitted the brasses to the pin by scraping, and bolt down the top slide bar with distance pieces between. Fix the regulator valve box (previously put together), the cylinder cocks and lubricators, the steam chest cover, and the back cylinder cover, making all joints with red-lead 'putty' between. The putty is a mixture of red and white lead, softened with boiled linseed oil. After covering the joint surface, a piece of soft hemp line is laid once or twice round, and the cover then put on. Portland cement or asbestos discs are also used.

The last stage of all is to carry away the parts to their permanent position, and bolt down the whole to its stone bed; connect up the steam and exhaust pipes, and get up steam.

We shall now conclude with one or two general points.

Templates and Jigs.—The former have been sufficiently explained in Figs. 253, 264, and 266. They are used very extensively in much repeated work, thus saving a great deal of time in marking off, and they take a variety of shapes. Jigs are an extension of the template principle. Instead of thin plates, castings of an inch or so in thickness are used, supplied with holes where needed, the object being to guide the drill to its proper place on the work without the necessity of lining-out at all. An example of the application of this principle to a cylinder cover is shewn at Y, Fig. 273. (*See Appendix II., p.* 820.)

Hobbing a Worm Wheel.—A cutter for forming *spur*-wheel teeth was given at Fig. 186, and a method of cutting *bevel* teeth at Fig. 262. Worm-wheel teeth can also be cut by first turning in the lathe a worm of the correct shape, and of good steel. This is then fluted to form a milling cutter, and is termed a *hob* in the workshop. The operation is then much the same as



Erection of Engine &c. *Fig. 273.*

Diagram of the New Standard Wire Gauge, shewing the Various Equivalents in Fractions of an Inch.

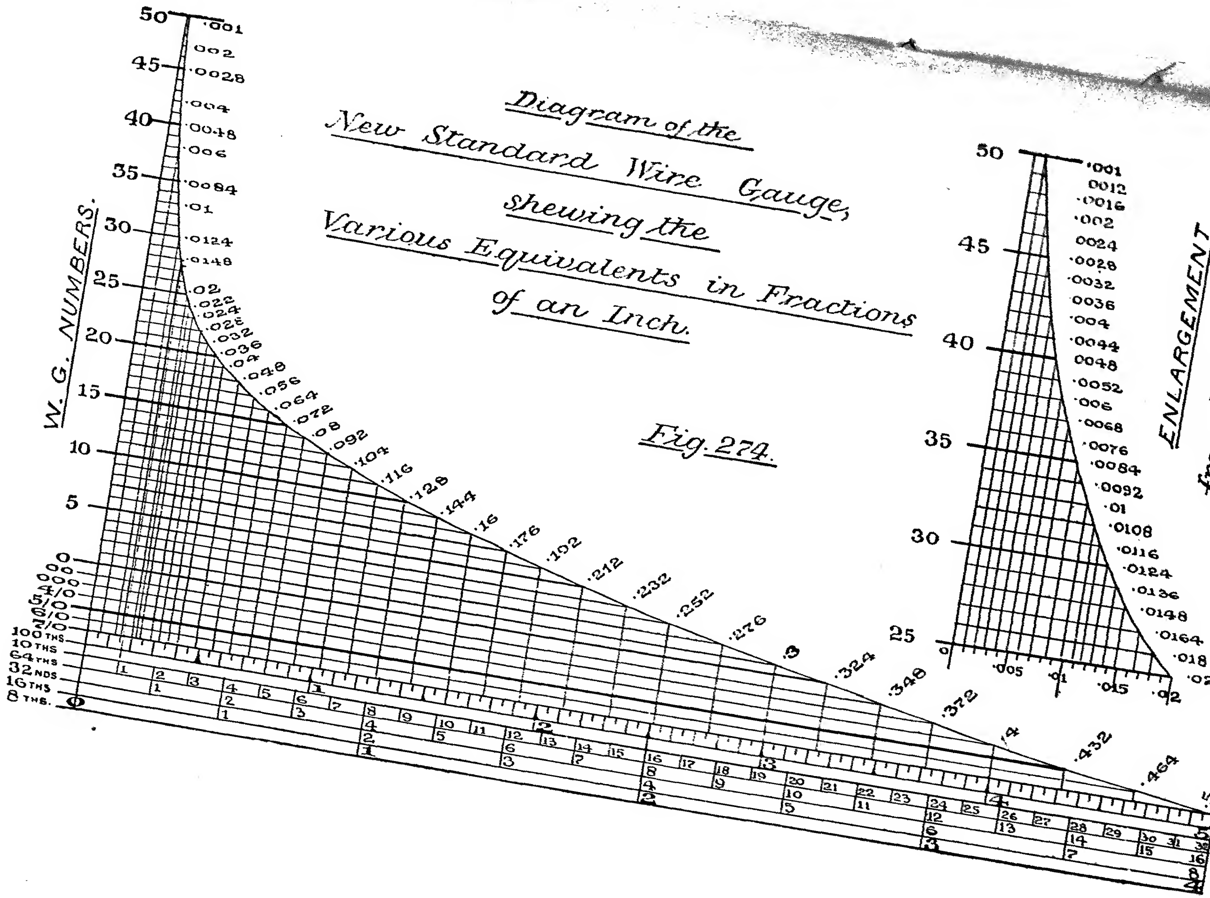


Fig. 274.

that described at page 58. The spaces are first cut on the angle with a spur-wheel cutter, and the finish given with the hob by placing both wheel and worm in position, as at Fig. 72, and rotating the latter on a milling spindle. (*See App. II., p.* 823.)

Dimensions.—In most workshops the inch is divided into vulgar fractions in the common way. But in dealing with work of great accuracy, or where small differences are to be represented, the above divisions have to be carried beyond sixteenths, and then become cumbersome. To avoid this difficulty, the decimal system of division has been used for a considerable period in a few shops, and has proved a great boon, being easily learnt by any workman, and its advantages greatly valued. We have spoken of *high and low gauges* for interchangeable work. Where these are used, the drawings are supplied with what are known as 'plus and minus' dimensions. Thus all shafts, pins, &c., are figured ·002″ larger than the size required, an inch pin being 1·002″; and holes are marked ·005″ larger than their pins, an inch pin requiring a hole 1·007″. There is an understood plus and minus allowance of ·002″ on both these dimensions, so that if a large pin and small hole come together, there will be a minimum clearance of ·001″, while a small pin in a large hole will have a maximum clearance of ·009″. For driving fit, the hole and shaft are figured the same, and the kind of fit noted. (*See Appendix II., p.* 825.)

It was long ago found advisable to fix the thickness of thin plates and the diameter of small wires by reference to a table of numbers, which received the name of the Birmingham Wire Gauge or B. W. G., and where each number had a corresponding dimension. This table was readjusted about the year 1885 and considerably extended, under the name of the New Standard Wire Gauge, and has been shewn diagrammatically in Fig. 274, the horizontal scale representing a length of half an inch, while the ordinates are referred to the numbers on the left hand. The actual gauge is represented at K, Fig. 273, being a steel plate provided with slots of the correct widths.

Split Pins.—Half-round wire split pins are made in fifteen different sizes, the largest being $\frac{7}{16}''$, $\frac{3}{8}''$, and $\frac{5}{16}''$, and the remainder numbered 1 to 12, corresponding with W. G. The

[illegible] as follows:

[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]

CHAPTER VII.

BOILER MAKING AND PLATE WORK.

WE now enter upon a division of practical engineering having no direct connection with any previously described processes, excepting only Metallurgy (Chapter III.). Boilers, Tanks, Girders, Ships, &c., are *built up* by bolting or riveting together plates previously manufactured at the *Rolling Mill*, intermediate connections being formed by '*section*' bars, rolled at the same place.

Materials. It is beyond our scope to give a detailed account of the production of wrought-iron or steel bars, angles, or plates, by rolling while hot. Wrought Iron is obtained from Cast Iron by *puddling*, as at p. 75, where the process was followed to the formation of bar iron of different qualities. The bars are now hot-rolled by passing between pairs of horizontal rollers, being supported on their way to or from which by a train of bearing rollers. Putting the bar through the mill a sufficient number of times (both crosswise and lengthwise), a **flat plate** is obtained; then sheared to rectangular shape. These rolls are too simple to need illustration. They are very powerful, being driven by a large engine, which is either itself reversible, or the rolls are supplied with a reversing clutch.

If '**Section**' **bars** are required, the material is first reduced to a convenient thickness, and then passed through a set of rolls capable of gradually decreasing the sectional area, while lengthening the bar. Thus any particular section may be obtained, the process consisting of '*cogging*,' or roughing-down the bar, and 'finishing,' or giving the true section. In Fig. 275, A represents a train of rolls for producing **L** or angle bar, *a* being the cogging set and *b* the finishing set; B shows rolls for plain or merchant bar; C those for **T** bar; and D for **H** (aitch) bar. The operations occur from left to right, the upper rolls being usually provided with discs riding on corresponding depressions in the lower roll, and thereby preserving the correct thickness of bar.

Iron plates retain the fibrous quality imparted to the bar, and are therefore much stronger in the direction of the fibre than across it. Owing to the secretion of cinder and scale between the layers during piling, the finished plate must be carefully examined for faults—(1) by eye, (2) by slinging from the four corners and tapping, when the dull, ashy portions may be detected by the non-vibration of sand sprinkled over the surface.

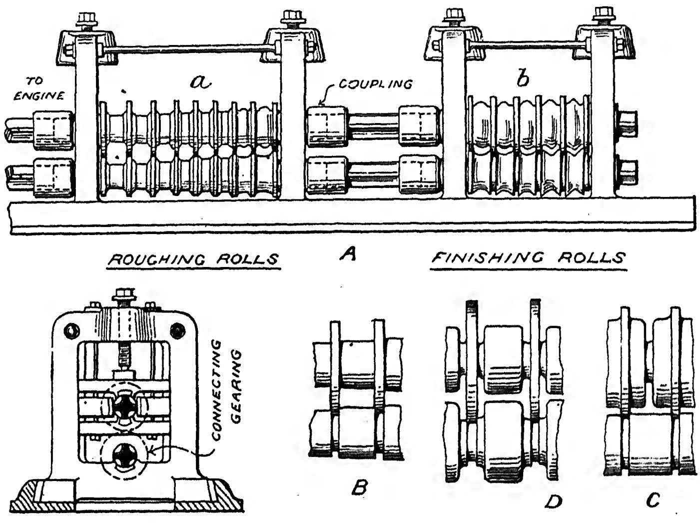


Fig. 275.

Rolling Mill for "Section" Bars.

Very bad plates are rejected, and the others placed in the scale according to quality, thus evolving the various degrees of 'best,' 'double best,' and 'treble best;' terms, however, by no means sufficiently definite. The Yorkshire irons are made with great care and a large expenditure of fuel, being also very carefully selected.

Steel Plates and Bars are rolled similarly. The ingots, obtained as at pp. 79 to 82, are, after casting, usually broken up, piled, and re-heated, though some authorities complain that this destroys the homogeneity for which steel plates are admired, and prefer to roll direct from the ingot. The slabs or ingots should

be well squeezed in both directions when made into plate. Steel plates are much more reliable now than when first introduced, it being clearly recognised that a certain amount of strength must be sacrificed to ductility. They are not, therefore, considerably stronger than iron, but much more homogeneous or even in structure, the particles being so thoroughly re-arranged when in the molten state. Iron plates, on the other hand, are very various in quality, even over one plate, because of the processes employed in obtaining fibre. A test strip, either for plate or bar, rarely gives an exact determination of the whole, while the contrary holds with steel. Steel plates are termed 'mild' because they have little more carbon than wrought iron plates; they have some 30 per cent. more strength than the latter, with one and a half to twice the elongation. The price of iron being also greater, it is not surprising that steel is the only material now used for plate work, excepting where continuous flame action (as in locomotive fire-boxes) renders Lowmoor iron or copper preferable. **Copper**, though more expensive, and losing its strength somewhat while hot, is an exceedingly good conductor of heat, and deteriorates less under the action of flame, lasting therefore longer than iron, while being more efficient.

Steel, then, in a mild, homogeneous form, is the material now generally used for all plate and angle work. Both steel and iron are received from the rolling mill under the following forms:—

Plates	——	H (Aitch) Bars	H
Angle Bars	L	Flat and Square Bars	▬
Tee Bars	T	Round Bars	●
Channel Bars	⊓		

Brands, qualities, and sizes of plates.—The qualities of iron have been mentioned at p. 76, 'common' being used for bridges and girders, and the remainder for boiler work. **Mild Steel** occurs in four qualities, thus:—

1. Ship and bridge quality.
2. Ordinary boiler quality.
3. Soft boiler quality.
4. Superior quality (to resist flame).

The brand **BT** [illegible] Board of Trade test [illegible] **B** [illegible] other brands representing [illegible] makers.

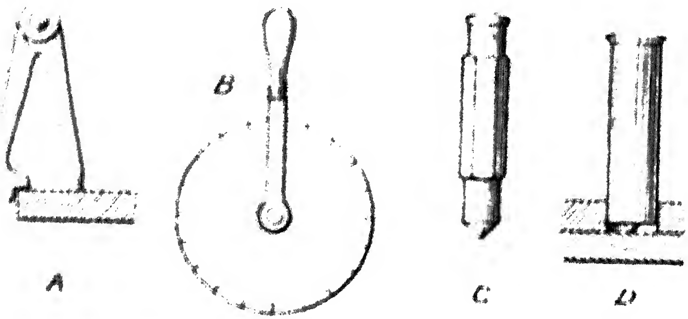
[illegible]

Maximum Sizes of Steel Plates [illegible]

Thickness	Length	Breadth	Area in Square Feet
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]

Rivets are prepared from round bar. If of iron, they should be of the very best quality—"Swedish," "Charcoal," or "Low Moor," and capable of standing re-heating without deterioration. Steel rivets have now almost superseded those of iron. The greater strength of the plates was of little value so long as the joint (the weakest part) remained much as before, but since the introduction of hydraulic riveters by Tweddell, and with care in re-heating, there is no objection to steel. The making of rivets is shewn at p. 99. (Also *Appendix V., p.* 970.)

Hand Tools, &c.—The boiler-maker and plater require somewhat different sets of tools. Both men must be able to



Marking Tools *Fig. 276.*

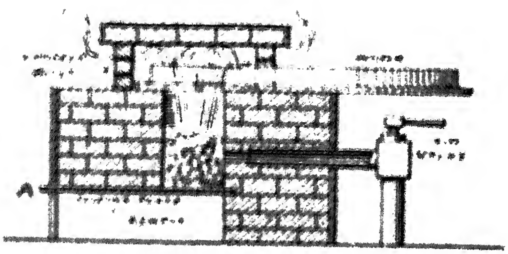
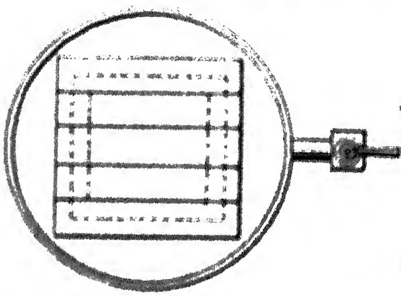


Fig. 277.



Boiler-Maker's Hearth for heating Angle Bars &c.

mark out their work, although in large shops a separate man is kept for the purpose. In the latter case the work may be further subdivided among template-makers, *platers*, *riveters*, and *angle-*

[illegible]

[illegible] **Marking off** [illegible] *[illegible]* [illegible] plates [illegible] there are two [illegible] other [illegible] plates [illegible]

The **Angle-iron Smith** [illegible] a welding [illegible] between the [illegible] His hearth [illegible] placed [illegible] the blast entering [illegible] the [illegible] with slabs of the same [illegible] cast iron [illegible] good welding heat obtained without difficulty. [illegible] required for heating plates; they are [illegible] but have a flatter [illegible] and a larger [illegible] of the plates. A [illegible] heating furnace [illegible]

The **Boiler-smith's tools** are [illegible] Chap. IV. In addition to the tools, Fig. [illegible] is required for finishing [illegible] both curved and straight in profile, and [illegible] Fig. [illegible], are also necessary. For tongs, [illegible] and [illegible] are useful [illegible] for lifting angle irons, [illegible] for flanges, and [illegible] for rivets.

Plates may be flanged, bent, or straightened by hand, large wooden hammers being then used, as at [illegible], Fig. [illegible], three of which are employed by as many men, who give equal consecutive blows; but these processes are done by machine where possible, and will be described later.

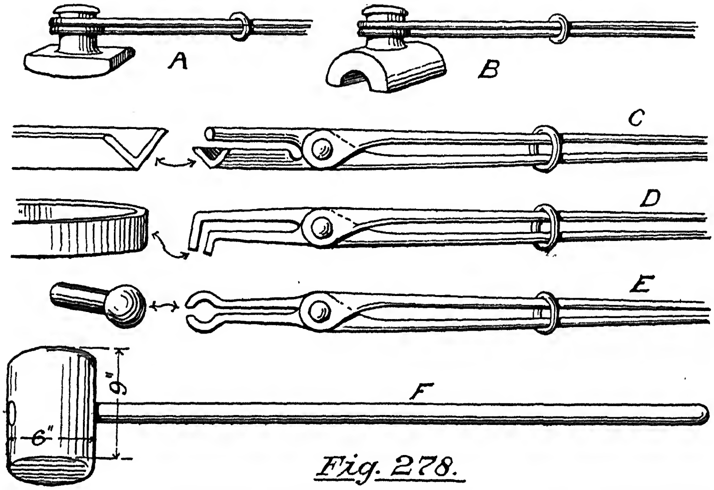


Fig. 278.

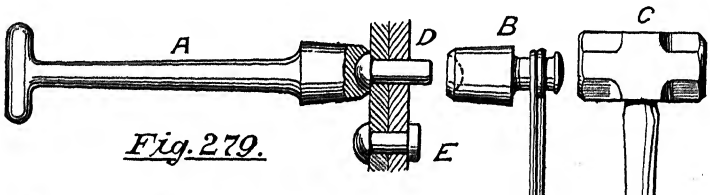


Fig. 279.

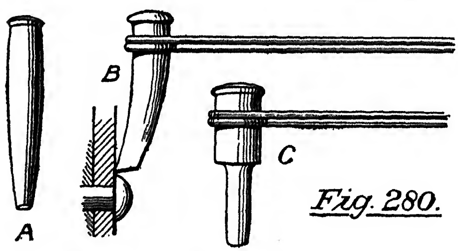


Fig. 280.

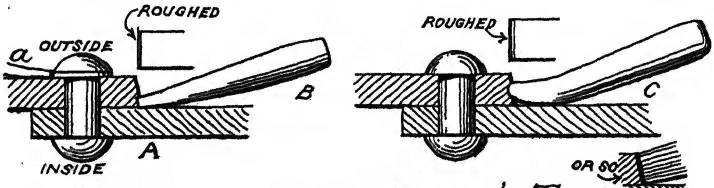


Fig. 281.

Boiler Maker's Tools.

The **Plater** requires three *chisels*—the flat chisel, Fig. 201, the cross-cut, as at Fig. 200, and another with curved profile for chipping the edges of manholes, &c. *Hammers* are of three kinds—the fitter's hammer, Fig. 199, the sledge hammer, and a riveting hammer with long head and small panes for places where the sledge or the portable riveter cannot be employed. A **Riveting Gang** consists of three men and a boy; the boy brings the red-hot rivet, which the leader inserts, as at D, Fig. 279; another man holds up the dolly, as at A; while the third man and leader give alternating blows until the cheese head E is formed. The leader then applies the cupping tool or snap B, while the striker gives two or three smart finishing blows with the sledge C. Work should be designed for machine-riveting wherever possible, as hand work can neither make the rivet completely fill the hole or compete in cost.*

Before riveting a seam, the plates, if punched or drilled separately, are brought into alignment by the podger and bolted in one or two places; then the *drift* at A, Fig. 280, may be applied and forced through by a hammer to clear out the holes. Though of undoubted advantage if used temperately, the drift is now banished from the best shops, plates being injuriously distressed by it when the holes are very untrue. When a joint is to be broken, the rivet-heads are chopped off by the set B, struck with a sledge, and the punch C applied to drive out the rivet.

Caulking is the process of making a boiler joint thoroughly staunch by burring up the plate edges with a blunt chisel or caulking tool. In Fig. 281, A is the section of a boiler joint, where the edge of the outer plate is bevelled at an inclination of 1 in 8. Striking the tool B with a hand hammer a burr is formed, and the rivet heads treated similarly, as at *a*. Severe caulking with sledge diminishes the grip of the rivet and frictional strength of the joint. To avoid this a fullering tool C is often used, but there is no objection to caulking if a large number of light blows be given. A Pneumatic Caulker will be described later. Caulking the rivets is not considered necessary if hydraulic riveting be properly applied. (*See p.* 322, also *Appendices II. and IV., pp.* 826 *and* 949.)

* See diagrams by Mr. Tweddell, prepared for his paper before the North-east Coast Institution of Engineers and Ship-builders, p. 321.

Punched *v.* Drilled Holes.—Formerly the holes were punched in a boiler plate before rolling the latter into cylindrical form, and alignment then obtained by very forcible use of the drift. The holes were marked by dipping the end of a short piece of brass tubing into white paint and transferring to the plate; the puncher could not therefore give great accuracy, and the plate needed considerable stretching when a pair of holes

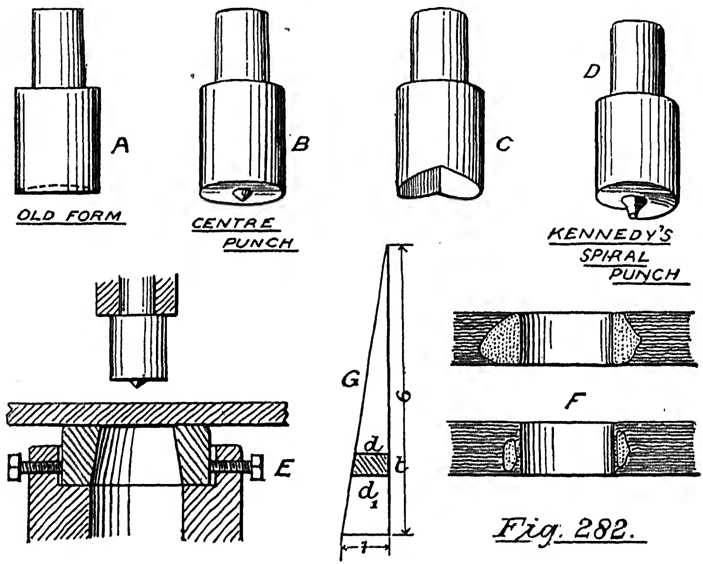


Fig. 282.

Punches.

made 'half moons.' Later the centre-pop replaced the white ring, and a 'centre' punch as at B, Fig. 282, was used in the machine, so that the hole could be punched with accuracy. The machine punches thus took the successive forms, A, B, C, and D. C was introduced to avoid distress of plate by giving a *gradual* shear, and D, Kennedy's spiral punch, still better carried out the idea of C, as proved by actual tests. The bolster is shewn at E, to support the plate while punching; and the size of hole (larger than the punch) may be found by construction at G, a triangle

being drawn with sides as 1 : 6. Then if d be diameter of punch, and t plate thickness, d_1 will be the size of hole in bolster, or

$$D = d + \frac{t}{6}.$$

The material removed from the plate is known as the 'punching,' or 'burr,' and during the operation a certain portion is compressed into the surrounding plate, thereby increasing its density and causing 'distress;' the clearance between punch and bolster hole is to prevent this, which it does partially. The distressed area is said to be small, and the distressment relievable by rimering, annealing, or both. Dr. Kirk's experiments in 1877 on the fracture of punched plates, shewed the crystalline or weak portion varying between the two limits at F, Fig. 282. All this was removed by subsequent annealing, heating to redness, and slowly cooling.

But the question was raised: if the plates require such treatment after punching, and alignment be not then obtainable unless punched *after* rolling (very difficult with machines as made), why not *drill* them at once and avoid annealing? There is no difficulty in drilling *after* bending, and further, the holes may be made through both thicknesses of plate at once, thus securing accuracy of position. Drilling 'in position' is therefore the present-day practice, and we are not aware of any workshop where punching is performed except for very thin plates, or for roughing out man-holes, &c. After drilling, the sharp edge is taken off by a countersinking tool, or rosebit, to prevent cutting action on the rivet, caused by expansion and contraction of the boiler.

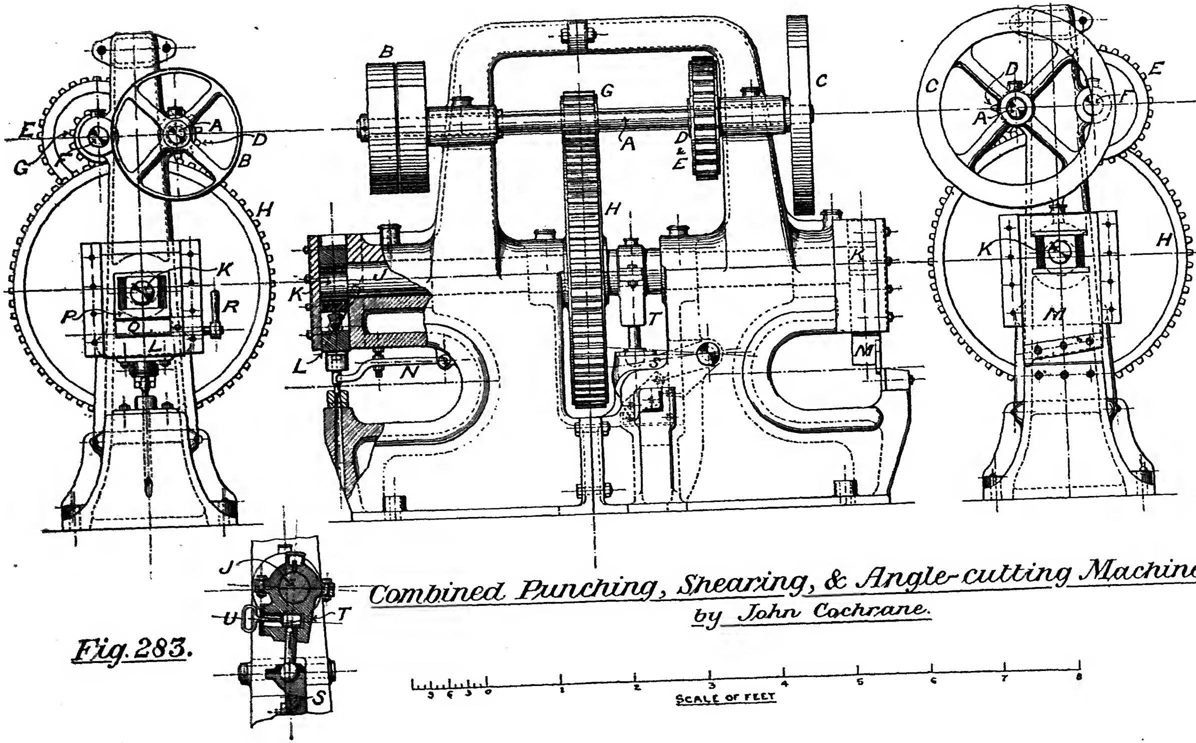
Shearing causes the same harm to the plate as punching, and the edges should always be planed afterwards.

D Cramps as at A, Fig. 219*b*, are required by boiler makers for temporarily fastening plates together, or for providing a hold when slinging.

Machine Tools, as explained in Chapter V., are daily gaining ground, to increase the output, while securing greater accuracy and cheaper production. As in the Fitting shop, they were at first driven entirely by belts from a line of shafting, but the intermittent demand renders hydraulic power more advan-

tageous. Mr. Tweddell advocated the almost universal application of hydraulics for plate work, and fully confirmed his advocacy of the system, especially where the power had to be taken about to various places in succession. In all shops *Riveting Machines* and *Flanging Presses* are now actuated by water pressure; so also may be *Punching* and *Shearing Machines*, though more often driven by shafting; while *Drilling*, usually performed by shaft power, has been successfully attacked by electricity and water pressure; portable hydraulic drills, under certain conditions, having proved both efficient and economical.

Punching and Shearing Machines.—It is customary to combine both operations in one machine, as a plate is seldom punched and sheared at the same time. Fig. 283 shews a good example of this tool, as made by Mr. John Cochrane, of Barrhead, capable of either punching, shearing, or angle cutting. A shaft A has fast and loose pullies at B, and fly wheel at C for overcoming variable resistance. The power passes, by pinion and wheel, D and E, to a second motion shaft F, and in like manner, by wheels G and H, to the main shaft J. The shaft J has eccentric pins KK formed upon its ends to give a vertical reciprocating motion to the slides L and M, the former carrying the punch, and the latter the shearing knife. Dies upon the pins KK prevent undue wear, and the fork N prevents the rising of the plate when the punch is withdrawn. The shearing knife always moves while the driving shafts revolve; but the punching slide L is driven from pin K through the hollow die P and a cam piece Q, the latter being connected to a handle R. When R is upright the downward motion of P is transferred to L: but if the handle be laid on its side, so also is the cam; P then moves freely without pressing upon L, and no punching occurs. Thus by changing position of R, the workman has ample time to set his plate, while the shafts still revolve. The dies are hard steel, and steel plates in slide M receive the wear. The angle-shearing knife is fastened to a rocking lever S, actuated from shaft J by an eccentric T, having ball and socket connection to the lever. Here, again, the withdrawal of a sliding piece U serves to stop the motion of the knife, which is necessary with bars, though not with plates.



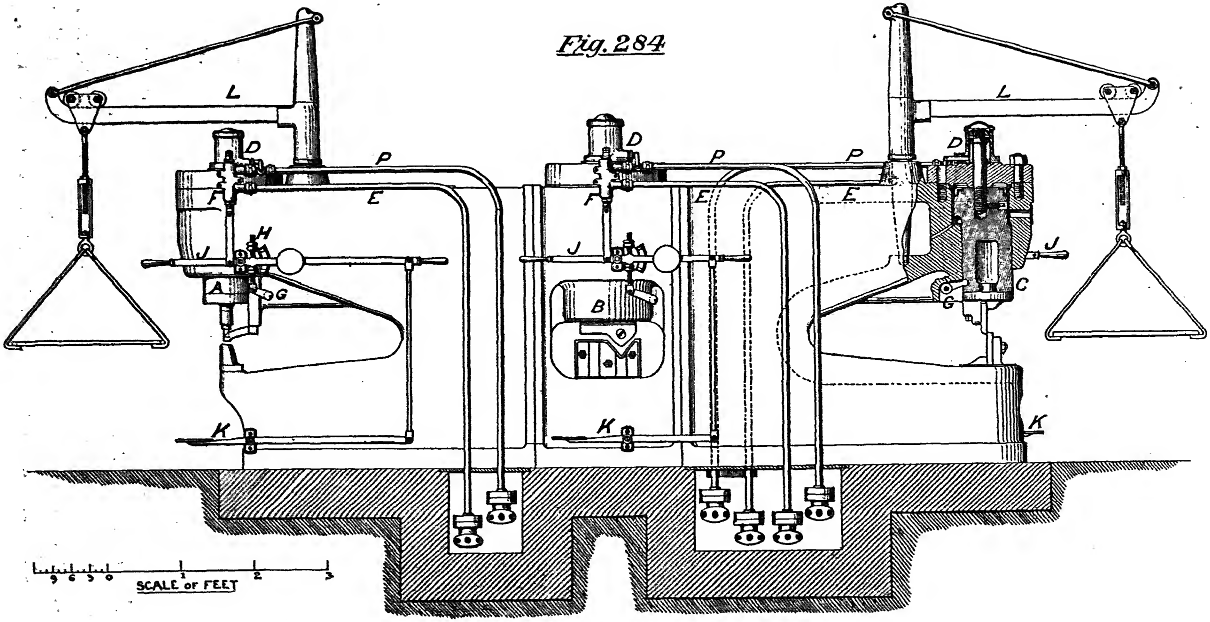
Combined Punching, Shearing, & Angle-cutting Machine.
by John Cochrane.

Fig. 283.

At Fig. 284 is shewn an **Hydraulic Punching and Shearing Machine,** designed by the late Mr. R. H. Tweddell, of Westminster, for performing the same operations as the foregoing by means of water pressure. In this example there is no reason why the three parts should be combined except to save floor space. A cylinder and ram are required for each operation: A for punching, B for angle-cutting, and C for shearing; there are also the lifting pistons at D D D. Water being supplied from the accumulator pumps at a pressure of 1500 or 2000 lbs. per sq. inch, two pipes are connected with each cylinder, one for 'pressure' and the other for 'exhaust,' marked P and E respectively. The valve boxes at F are supplied with piston valves (worked from hand and foot levers J and K) to control the supply and exhaust; but a *constant* pressure, on the pistons D D, causes the rams to rise when water is exhausted from the main cylinder. A small lever G, moved by ram C when at the end of its down stroke, is connected to a screwed rod H, having adjustable discs, which restore the levers J nd K to the horizontal position, stopping the water supply and the movement of ram C: this is known as cut-off gear. Two overhanging cranes L, L, support the plates while being operated on.

The **Multiple Punching** *or* **Shearing Machine** in Fig. 284*a*, on Tweddell's system, has been designed to prepare plates required in forming wrought-iron pipes for conveying water or oil across country, and known as 'pipe lines;' it is also useful for ships' funnels and masts, and for girder work generally. A shearing blade or row of punches can be attached at will; the latter being shewn in operation at A. The punches are set alternately low and high, so that the punching resistance commences gradually, and they are attached to a beam B capable of vertical movement. Downward motion is obtained by a leftward travel of bar C, whose lower rollers press upon beam B, while the upper ones re-act upon inclined planes D, D, D, fastened to the framing. The working ram E (see enlarged section) moves bar C; water entering the cylinder F from behind, and connection between C and E being made with a toggle G, to allow for vertical travel. H is the valve box with piston-valve moved by lever J, and the cut-off is effected automatically by the bell crank K, as

Fig. 284



Hydraulic Punching, Shearing, & Angle-cropping Machine.
(TWEDDELL'S SYSTEM)

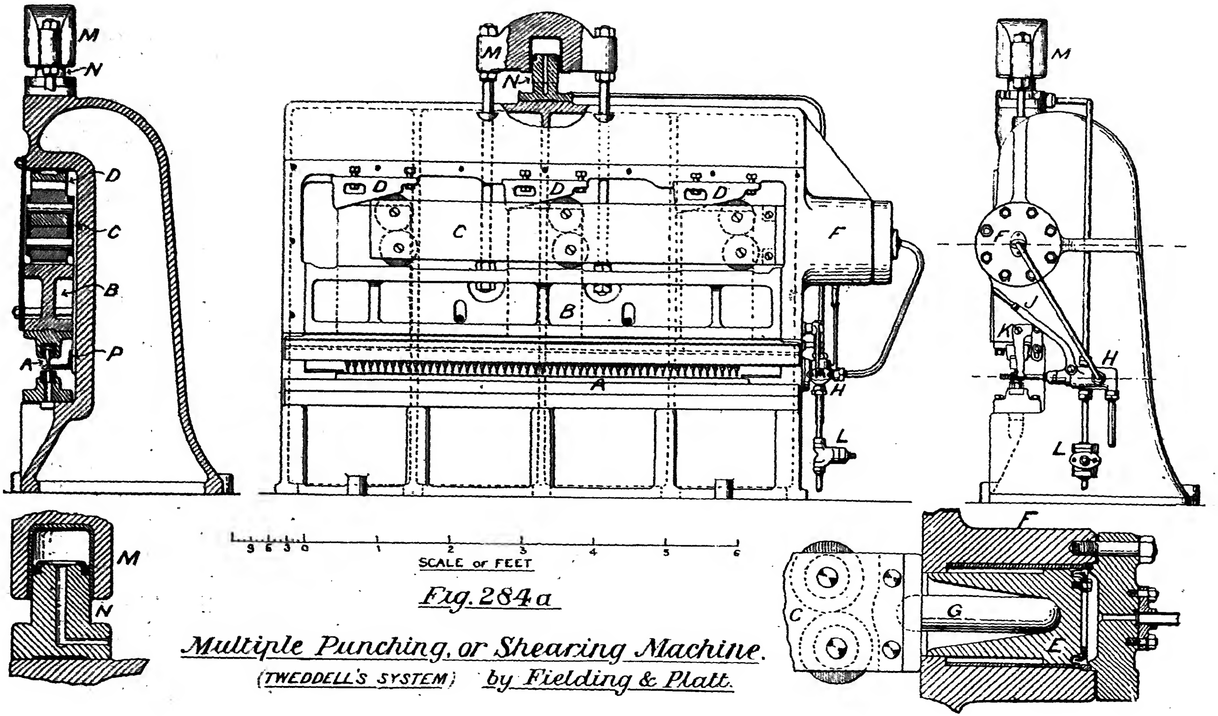


Fig. 284 a

Multiple Punching, or Shearing Machine.
(TWEDDELL'S SYSTEM) by Fielding & Platt.

previously described. A fixed ram N on the top of the framing, has a cylinder M in the form of a girder, to which a constant water pressure is supplied, and the girder is connected by bolts to the beam B, so that a rise of the latter takes place whenever the main cylinder is opened to exhaust. The angle bar P prevents the plate from lifting, and L is a stop valve.

A **Plate-edge Planing Machine** is shewn at Fig. 285, having a long table A, upon which the plate is clamped by screws B B. The tool C is fixed in a cylindrical box, provided with handle D resting on stops, so that direction of tool point may be reversed at either end of cut, shewn by the arc E; this is performed by the workman, who travels on a platform F attached to the saddle V. The latter has a hand-wheel and screw G to set the tool, while the wheel H, turned by hand, gives vertical feed. The saddle is traversed by screw J, driven from the countershaft K by gearing: while K is provided with fast pullies M, N, and loose pullies L L L. When the forks are in the position shewn, no work is done, but if the straps (crossed and open) be moved to the right the saddle will travel to the left and *vice versâ*. Reversal is automatically effected by projections P P striking the stops Q Q at either end of the stroke alternately, thus moving the straps, decision being given by the weight R, which causes a pressure between the rollers at S. The mid position is fixed by stops T: and the standards are so arranged at U that they *overhang* the work, thus allowing the admission of any length of plate. One setting may serve for several plates.

A Band Sawing Machine is a very useful tool in a boiler shop for cutting out plates of intricate shape, while straight plates, too thick to be sheared or punched, are cut by a *circular saw* when necessary. As these are so well-known in their wood-working capacity, diagrams have been thought unnecessary.

Plate-Bending Rolls, in their most common form, are shewn in Figs. 286 and 287, the rollers being supported horizontally. These are the design of Mr. John Cochrane, of Barrhead. The lower rolls A A revolve in fixed bearings, while those of the upper roll B are lifted or lowered by the screw C, the worm wheel D acting as a nut, while the worm is turned by the spoked wheel E. A A are the driving rolls, and the gearing is very

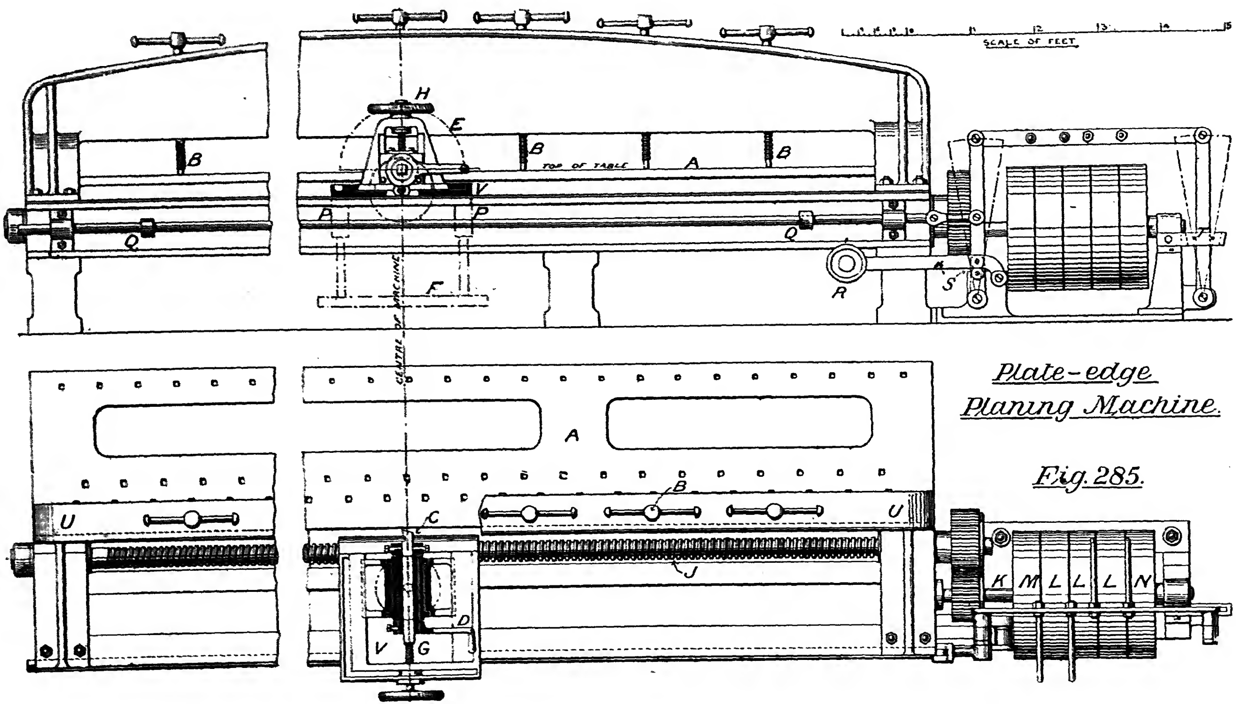


Plate-edge
Planing Machine.

Fig. 285.

Fig. 286.

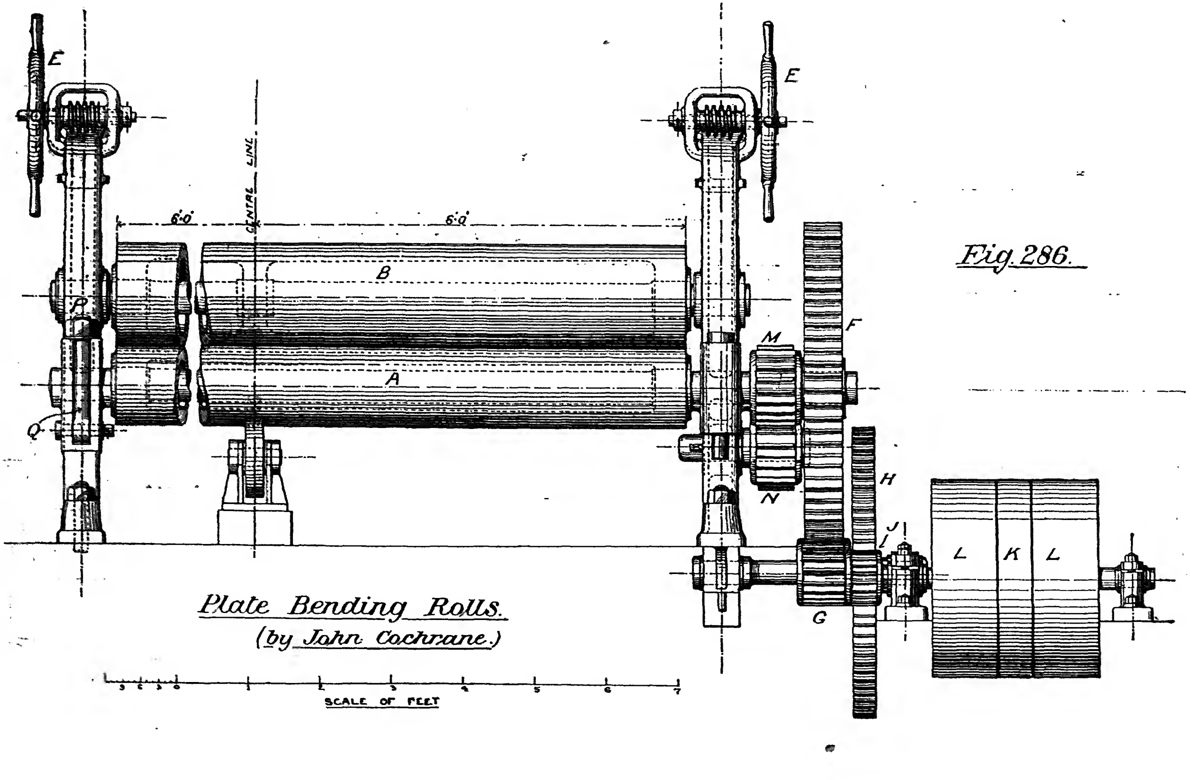
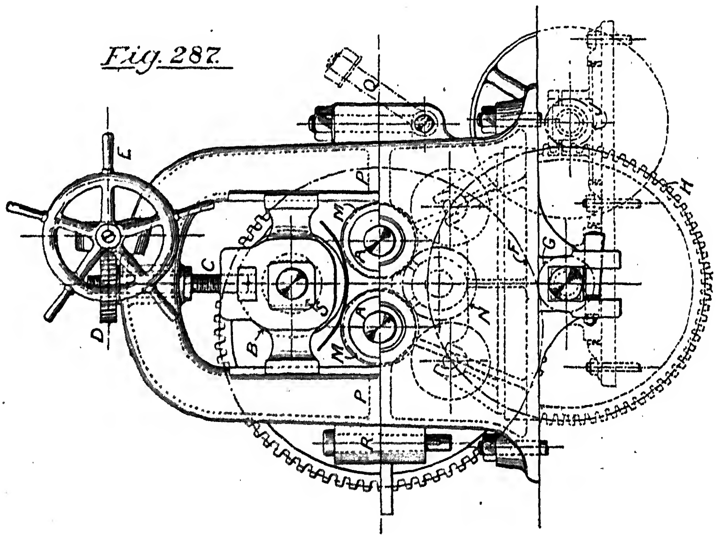


Plate Bending Rolls.
(by John Cochrane.)

powerful, consisting of wheels and pinions F G and H J, the last being on the driving shaft, while M M N connect the rolls. The pullies are driven by crossed and open straps, to obtain reversal, K being the fast, and L L the loose pullies, so that either strap may be put upon K alternately by a foot or hand lever attached to the forks (not shewn). The plate to be bent is placed upon the rolls A A, B lowered till a grip is obtained, and the machine set in motion. When the plate has been drawn nearly through, the

Fig. 287.



End view of Plate Rolls.

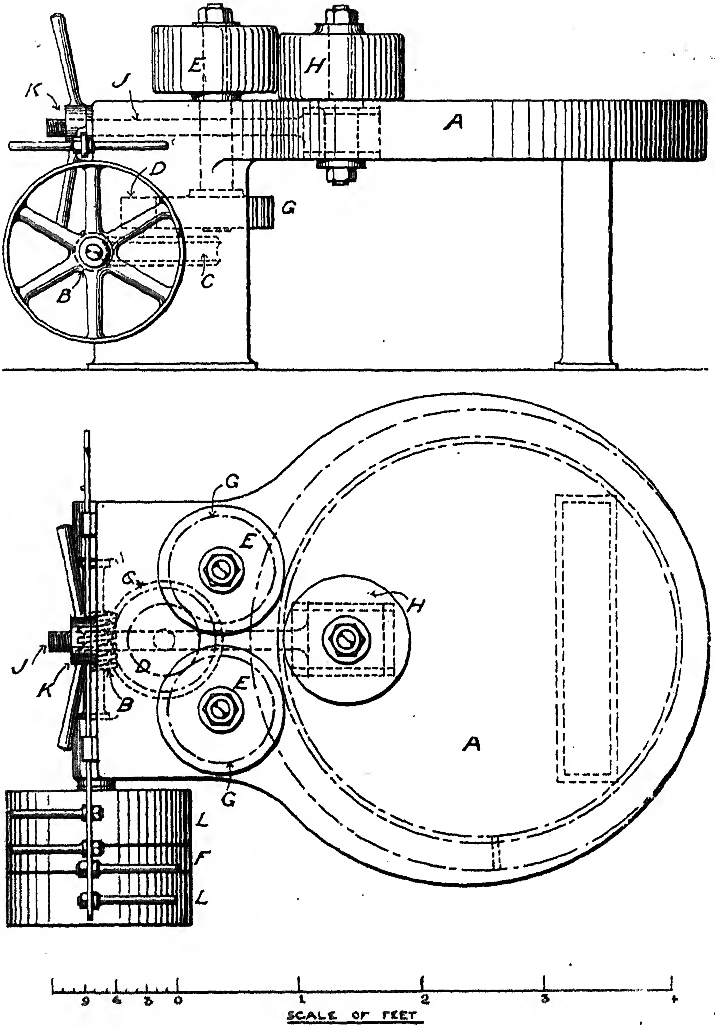
machine is stopped, and the wheels E E given a slight advance, the rolls then reversed, and the plate brought back, and these operations repeated until B is depressed enough to give the necessary curvature. When the plate is bent into an entire circle it cannot be released at the front; so the top of the standard is made separate at P P, and the bolt Q turned down as shewn dotted, when portion P P may be swung round horizontally upon pin R, leaving the bush S upon the roller B. The plate may then be withdrawn. It should be noticed that the sides of the bushes are curved in plan to radii from the pin R.

Vertical rolls are often used for long, heavy plates, and are said to be less expensive in operation, while giving truer finish to the end of the bent plate. This last is the principal difficulty with all rolls, the entering edge, to six inches deep, being always set by bending while hot with wooden hammers. Except for this, the plates are never heated for rolling, even up to $1\frac{1}{4}$ inches in thickness, for in such cases the radius is proportionately larger. The weight of plate is eliminated by the vertical method, with less fear of obliquity of curvature. Long rolls are often slightly bellied at the centre, to take up spring. For the heavier plates an *hydraulic bender*, introduced by Mr. Tweddell, seems very likely to supersede rolls. It finishes the plates to a truer circle from end to end, and there is no limit to plate thickness, or risk of fracture by too rapid feed. Butt strips can also be bent truly to boiler curve. The tool is similar in design to the multiple punch in Fig. 284*a*, but the girders are placed vertically, and suitable dies inserted instead of the row of punches.

Plate-straightening rolls are similar in construction, but there are some four rollers at top, pressed down simultaneously by connected screws, upon three rollers at bottom, and the plate is passed through and through till truly plane.

Rolls for Section Bars (Fig. 288) have their axes vertical, and are placed upon a table A, which is sometimes conveniently set flush with the ground, with a pit for the gearing. They are driven by the usual fast and loose pullies F and L with crossed and open straps for reversal. These actuate a worm and worm wheel, B and C, and a spur pinion D on the axis of C gears with wheels G G on the roller shafts. Thus E E are the driving rollers, and H the bending roller, with a screw J to bring its bearing closer to the rollers E E, effected by turning the nut K. A ring or angle bar is shewn bent to a circle with an outward flange—an inward-flanged ring being obtained by turning all the rollers upside down, and other sections by special rollers. Finally the ring is removed and welded with a glut-piece.

√ **Flanging Presses.**—It being always advisable to diminish the number of joints in a boiler, the end plate is usually flanged or bent over at the edge to form a ledge for the shell-plate, while stiffening itself considerably.



Angle-bar Bending Rolls.

Fig. 288.

Plates were formerly flanged entirely by hand, being moulded on cast-iron forms by blows from wooden hammers, as at Fig. 278. This method was slow and expensive, and two kinds of hydraulic presses are now used, (1) the 'Piedbœuf'* press for flanging at one heat, a very effective tool, but requiring separate dies for every separate kind of work; (2) the universal, or three-ram flanging machine, invented by Messrs. Tweddell, Platt, Fielding, & Boyd, and capable of either progressive or single-heat flanging. We will take these tools in order.

The **'Piedbœuf' Flanging Press**, on Tweddell's system, is shewn at Fig. 289. It consists of an hydraulic cylinder A containing a ram B, which may be raised on the admission of water pressure, thus lifting the table C, on which is placed the lower die D. A girder E carries the upper die R, being supported by guides F F, provided with nuts for the adjustment of E. The girder G supports the central cylinder A, and four cylinders, H H, containing the 'vice' rams J; and as it is necessary to move the cylinders H H to varying distances from the centre, the pressure (or exhaust) pipes are trained through three-quarters of a revolution between their connections at the pipe circuit K K and those of the cylinder, so that the pipe is not strained materially when the positions of H H are changed; in addition there are sheaths L L to prevent snapping at the unions. The valve-box M has two hand levers; N for controlling the vice rams, and P for the flanging ram. The two dies are shewn ready for flanging a tube plate Q, which has been made red-hot and laid on the lower die D. The vice rams are first advanced until the plate is held against the upper die R; then the flanging ram B is slowly raised and the plate made to assume the dotted form. The levers being reversed, the plate may be withdrawn. These presses are made large and powerful, but are not used for plates beyond 7 feet diameter, and rarely up to that.

Universal Flanging Press (Tweddell's system).—This very useful machine is shewn at Fig. 290. There are two vertical rams, A acting as vice ram and known as the 'elephant's foot,' and B for flanging the plate on what is known as the 'progressive system.' A third and horizontal ram C gives the finish, and a

* Pronounced *peeaybuf*.

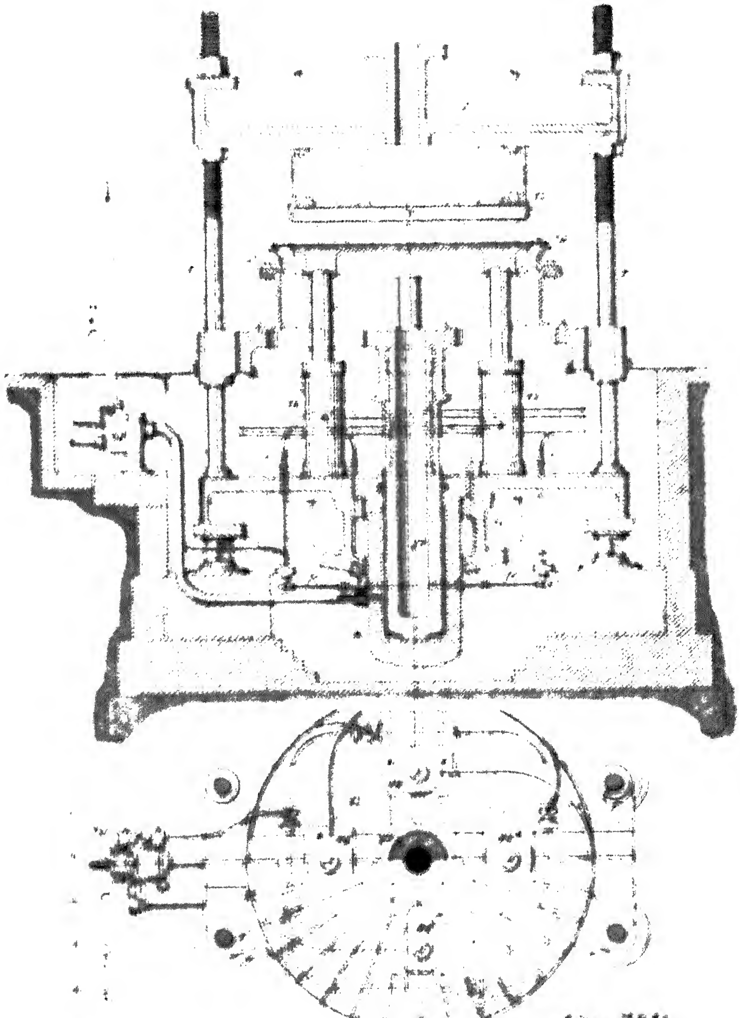
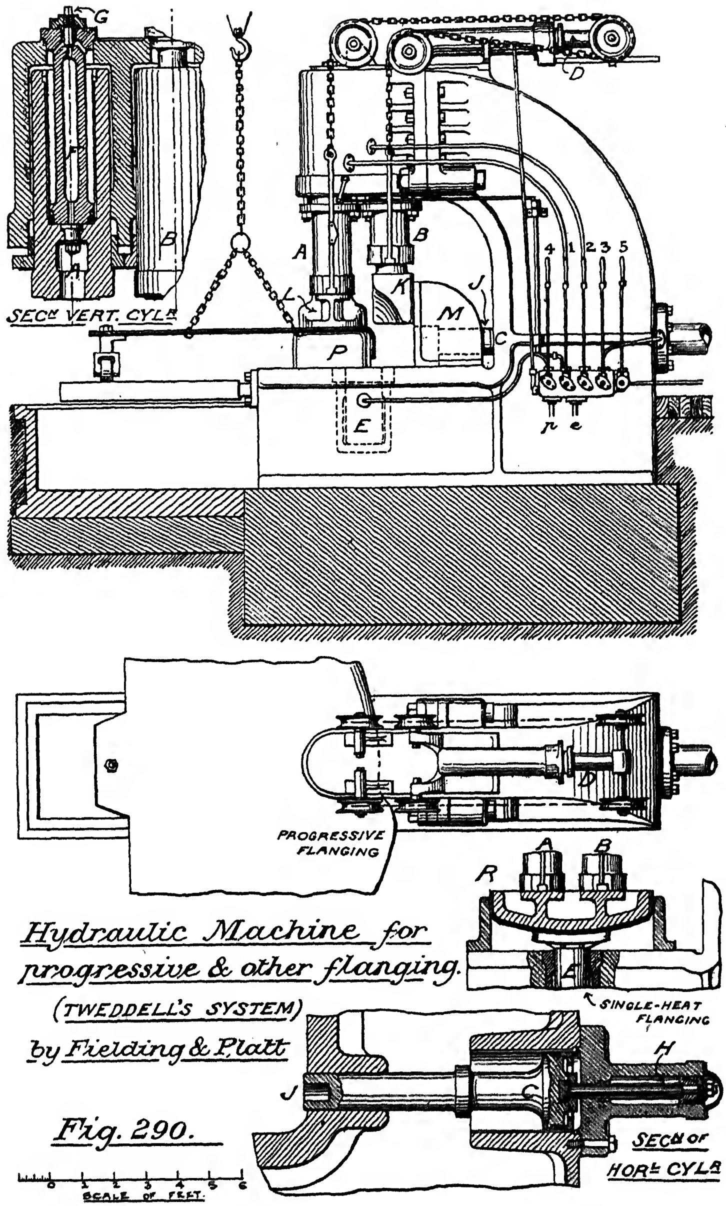


Fig. 280

Hydraulic Flanging Press.
Pedestal pattern.



Hydraulic Machine for progressive & other flanging.

(TWEDDELL'S SYSTEM)

by Fielding & Platt

Fig. 290.

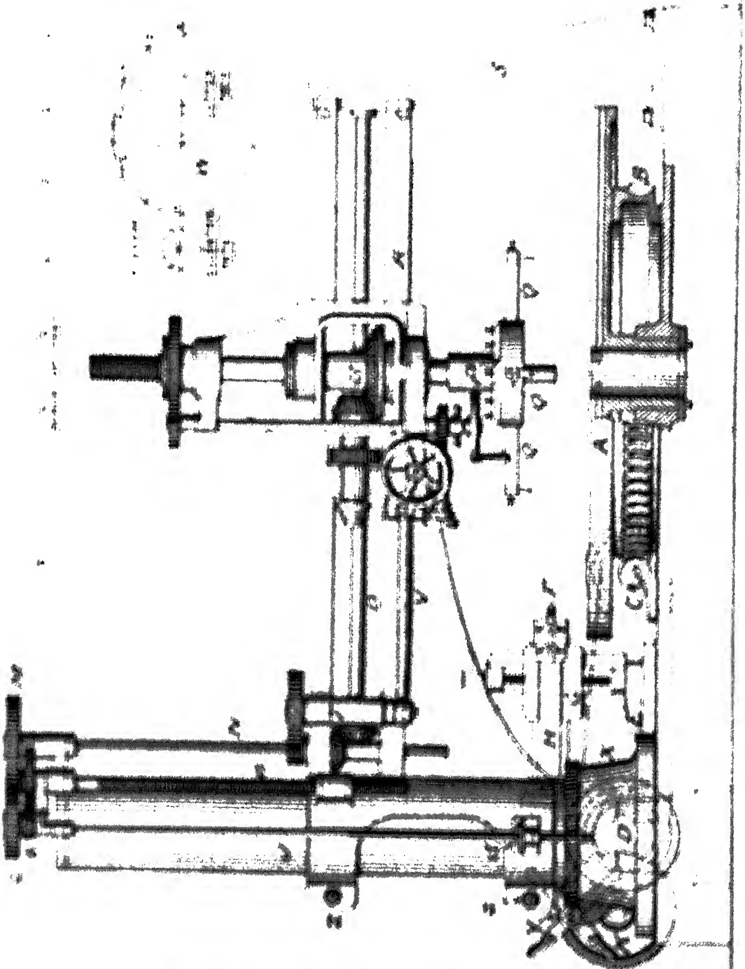
fourth ram D raises and balances the vertical rams A and B, having a constant pressure supply; so that the rams A and B only rise when opened to exhaust, one or other, or both. Yet a fifth ram E serves as vice during single-heat flanging. Referring to the enlarged sections, the ram A is seen to be hollow, riding upon a smaller fixed ram F. Ordinarily the water only enters the annular space round the small ram, but on releasing plug G it passes down the centre tube and then exerts a pressure on the whole area of the large ram, a variable power being thus obtained The horizontal ram C is of piston form with a tubular continuation to a smaller piston H, upon which there is a constant pressure, so the return is effected when C is opened to exhaust. Any special forms of dies may be applied at J, K, and L, and the guide bracket M is removable. The valve-box has five levers, each working both pressure and exhaust, 1 for ram A, 2 for ram B, 3 for ram C, 4 for ram E, and 5 for an hydraulic crane to lift the plates (see A, Plate XIV.). A plate N is being flanged on the progressive method. It is slewed by crane, laid on a curved hearth (B Plate, XIV.), and heated for a few feet along its edge, then transferred to the block P and flanged as described, rams A, B, C being applied in succession. This is done foot by foot until the heated portion is all flanged; a new heat then taken, and the work continued as before. When flanging with complete dies, the upper die is fastened to the rams A and B, as shewn at R, and the lower die placed on the table. The hot plate being laid on the lower die, the vice ram E is first raised and the upper rams then lowered; the flanging pressure is therefore the difference of that upon the lower and upper rams. Any kind of flanging can be performed by this machine by using suitable dies.

Drilling Machines, for boiler work, vary greatly in their construction. Except for the Radial machine they are all designed to drill 'in position,' and their form depends on the kind of work to be done. When possible they are made expeditious by the use of more than one drilling head, a necessity in view of the large number of holes to be drilled.

Radial Drill.—This has been already described at p. 167. Opinions differ regarding the best construction, but in almost any form it is an extremely useful tool for boilermakers. An inter-

esting example is shewn in Fig. 291, designed by Messrs. Geo. Booth & Co. for performing a variety of operations. The circular table A, provided with worm wheel B, may be revolved whenever the worm shaft C is connected to the driving shaft D by belt; at other times it is stationary. A bracket E, fixed upon the bed of the machine, carries a tool F through the medium of the two slides G and H, each provided with hand wheel and screw, thus giving adjustment in both directions. When, therefore, a boiler end plate is fastened to the table through temporary rivet holes, and the worm gear connected up, the tool F serves to turn the outer edge, the usual back gear being seen at K. The power further passes through mitre wheels and vertical shaft within the pillar to the spur wheels L M, and thence through shafts N and O to the drill spindle, the feed motions being as previously described. The simple drilling done on this machine is the taking out of tube holes in the manner shewn at B, Fig. 169; but large flue holes are made by using the head P and three cutter bars Q Q held by set screws with removable cutters, forming in fact a large pin drill. In all cases a hole is first drilled in the plate to receive the 'pin' and steady the cutter, and the radial arm R being long may be fixed to the bracket S when doing heavy work. But the most interesting feature to the student is the method by which large oval holes may be formed, such as those required as man-holes. A short vertical shaft T is connected to the driving shaft N by gearing of 2 to 1, the same ratio as that of the bevel gear at U. At the lower end of T is an eccentric stud adjustable within certain limits, and a rod V connects this with the saddle. The shaft T making its revolution in the same time as the drill spindle, an inspection of the diagram at W shews that the cutter will be compelled, by the movement of the saddle, to mark out a true ellipse instead of the circle it commenced with, which will be understood by comparing the numbers; of course only *one* cutter bar can be used. The tube J may be turned round within the base X, for fine adjustment, by the worm gear at Y, but the position of the arm R is first roughly obtained by releasing the bolts Z Z. The lifting is effected by the screw *a*, driven from the central vertical shaft by spur wheels at *b*, reversed or put out of gear at will by the handle *d* moved horizontally. This machine

Fig. 291



Radial Drilling Machine.

for cutting Flue holes, Chest, Manholes

Tube holes, etc.

(By Geo. Booth & Co.)

therefore can perform no fewer than four operations—flue-hole cutting, oval manhole cutting, tube drilling or other single drilling, and boiler-end turning.

Although the foregoing is a very useful, it is by no means a usual tool. The rotating table is more often placed on a bed by itself, constituting a *vertical face lathe*/

Drilling in Position.—The plates of a cylindrical boiler being prepared and temporarily connected, the rivet holes are drilled right through the several plate thicknesses. If stationary machines are employed they must be supplied with a cradle or bed on which to lay the boiler, so that the latter may be turned round on its axis, and thus present all portions of its surface at various times to the drill. Obviously there are two principal ways in which the axis may be placed, vertically and horizontally, the latter being used for large marine boilers, while the former is advantageous when drilling locomotive or Lancashire boilers, though it has also been employed for marine work.

Drills with Boiler Axis vertical.—Fig. 292 illustrates this type of drill: and its individual application (the drilling of rivet holes in the flanges of boiler flues), will first be described. The machine is the design and patent of Messrs. Geo. Booth & Co., and is very ingenious throughout. The flue is bolted, with its axis vertical and central, upon the circular table A, and a handwheel B, being connected to the table by bevel gear C and worm gear D, serves as a dividing plate, its revolutions being counted to turn the flue through any fraction of its circumference between each operation. The saddles E F ride upon vertical standards G H, and contain horizontal slides J K, for adjustment to various diameters. *l* is the driving cone, and power is taken from horizontal shaft L by mitre gear to the vertical shafts M and N: from these the various motions are obtained. Thus the spur gear and mitre gear at O and P give motion to the horizontal spindles Q R, and from thence by mitre gear to the vertical spindles S T, which turn the drills U U and V V by spur gear. The vertical movement of the saddles is given by hand or power. When by power, a worm on shaft N gears with worm wheel W, which actuates a second worm and wheel at X, connected with the screw Y by mitre gear. The mitre wheel on Y rotates within a boss cast

Boiler Flue Drilling Machine.

(by Geo. Booth & Co.)

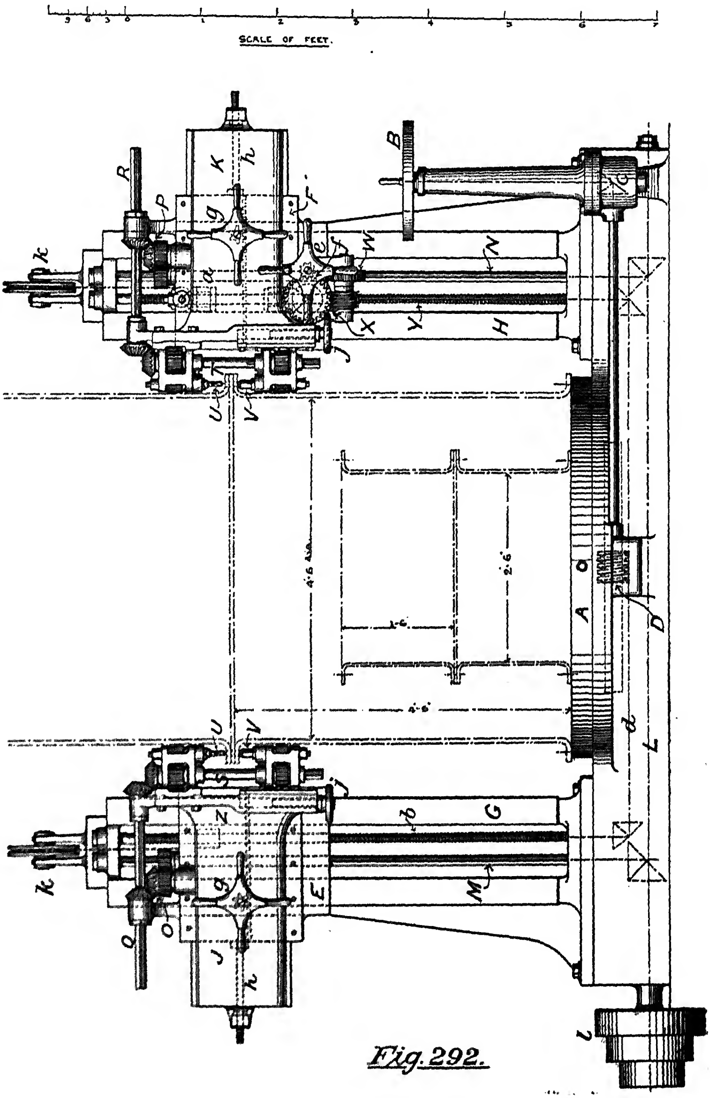


Fig. 292.

on the saddle, and has a plain hole, the connection with Y being by key only. There are two nuts Z and *a* upon the saddles, and the screws *b* and Y move simultaneously on account of their union by horizontal shaft at *d*. When, therefore, the driving shaft L is rotated in its proper direction, so also are the drills U U and V V, and a downward feed given to the saddle, as described. The raising or setting of the saddle involves hand gear, the capstan *e* turning the screws through pinion and spur wheel, and the mitre gear before mentioned: but although the spur wheel is keyed to its shaft, the worm wheel X is not thus secured, and is only in gear with the screw Y when *clamped* to the wheel *ff* while the nut *a* is carried in a socket, and is adjustable by mitre gear to alter the relative heights of the saddles. Horizontal adjustment is made by turning the capstans *g g*, each of which moves a pinion within a rack, and the bolts *h h* serve as adjustable stops. The drills themselves are worthy of notice. The upper ones, U U, are of the twist shape, but have a conical shoulder at the top, forming a countersinking bit. The lower drills V V are for countersinking only, and their feed, upward or downward, is obtained by hand wheels and screws *j j*. The saddles, somewhat loaded with all this gear, are coupled to chains passing over pullies *k k* to balance-weights behind. In drilling a flue fixed upon the rotating table, the saddles are raised by hand to approximate height, and advanced horizontally by the capstans *g g;* then the stops *h h* are set. The strap fork is moved on the countershaft and the drills rotated, while the feed wheel at X is clamped in gear. The hole being drilled to proper depth and countersunk, the feed is unclamped and the saddle raised to allow the bottom countersinking to be done by hand feed *j j*. Withdrawing the tools V V, the dividing wheel B is operated to turn the flue by the amount of the rivet pitch, and the next pair of holes drilled as before.

Shells of Locomotive boilers are drilled by machines similar in general build to that just described. A longer bed is needed, that the standards G and H may be advanced or separated by a tommy-bar applied to pinion and rack. An internal dog-chuck on the face plate grips the shell, and the dividing gear remains the same. The saddles are materially altered, being similar to those of the radial drill, Fig. 291, excepting that vertical screws

are applied instead of a rack. The drill spindle therefore lies horizontally, and might be represented by G and H, but the screw feed on its other end replaces slides J and K. Some makers withdraw the drill by power, using a quicker speed.

The larger shells of Lancashire boilers may be drilled similarly, but are often slung vertically by travelling crane, and held against a pair of vertical standards, which support the drill spindle at a fixed height. Such a method is, however, less capable of rapid and accurate adjustment. If there are two internal and two external pillars, the holes may be drilled and countersunk on both sides at one operation.

Marine boilers are sometimes drilled with axis vertical, on a rotating table as in Fig. 292, but usually are either laid horizontally, or a portable drill is applied.

Drills with Boiler Axis horizontal (Figs. 293 and 294). Plate XIII represents a machine for drilling the shells of Marine Boilers while laid horizontally. It is designed and made by Messrs. Hulse & Co. The boiler is placed upon a cradle consisting of four disc rollers A A A A, which can be turned by power applied to the worm shafts B B, so as to bring any portion of the shell circumference in front of the drill. The drill standards C C, carrying the saddles D D, may be moved to various positions along the slide bed E, and may also be adjusted, by turning on the hinges F F, so as to lie tangentially to the boiler, a condition *obtained* by the hand wheel and screw at G, and *tested* by the fork H, each of whose prongs should just touch the shell. There are fast and loose pulleys at I, giving power through spur wheels K to the principal shaft L, which forms a hinge pin for the standards. Within the standard base, mitre gear connects L with vertical shaft M, and from thence to drill spindle N through the spur gear O and mitre gear P. The feed screw takes its motion from the shaft at Q, through mitre gear Q, cone pulleys R, worm and wheel S, and mitre wheels T, and the saddle may be raised or lowered by the hand wheel U, the screw being turned as in Fig. 291. The hand wheels V V act upon a vertical shaft through worm gear, and thus turn a pinion within a rack on the inside of the bed for adjusting the horizontal position of the standards. The shaft L, besides driving the drills, also rotates the rollers of

[illegible] and the [illegible] shaft [illegible] the shaft [illegible] [illegible] gear at [illegible] [illegible] and the [illegible] at [illegible] cutter [illegible] according [illegible] whether [illegible] a central position [illegible] adjusting the standards [illegible] as usual, the outer [illegible] the [illegible] Fig. [illegible], while the [illegible] are [illegible] for the internal countersink and changed at each drilling. When two holes are drilled at once, the [illegible] one being being operated, the rotating gear must act as one.

[illegible] the shell. The front and back of the boiler, so far as the seams are concerned, are drilled by a machine of similar appearance, but the standards are exactly vertical instead of being hinged, and the cradle is turned through a quarter circle [illegible] that is, the front of the boiler would be seen in Fig. [illegible], while the side view would appear at Fig. [illegible]. When the lower seam has been drilled, the boiler is rotated through half a circle until the upper seam comes into position, the other holes having been previously taken out under the Radial Drill. (See p. [illegible].)

Portable Drill. — This machine being much less used than formerly, we shall only briefly notice it. It exists under two types: the bracket or standard type, bolted to the plate through holes already drilled by it, and the [illegible] type, well represented by Rolland's Drill in Fig. [illegible]. In either case the driving gear (by rope) is similar. Referring to the figure, [illegible] is the driving shaft, and the cord is held taut by weight [illegible], while the power is taken off by pulley [illegible]. An endless catgut band drives the drill spindles in opposite directions, and the feed is given by worm gear. Being clamped within the boiler, two opposite holes are drilled at once, the one pressure forming a reaction to the other. The feed is supplied either automatically or by hand, and the machine is capable of drilling shells 9 ft. in diameter. The safety clamps avoid accidents in case of breakage of the rope.*

Multiple Drill. — In addition to the machines already described, there is one, the Multiple Drill, which has always

* See previous remarks on portable hydraulic drills, page [illegible].

PLATE XIII.

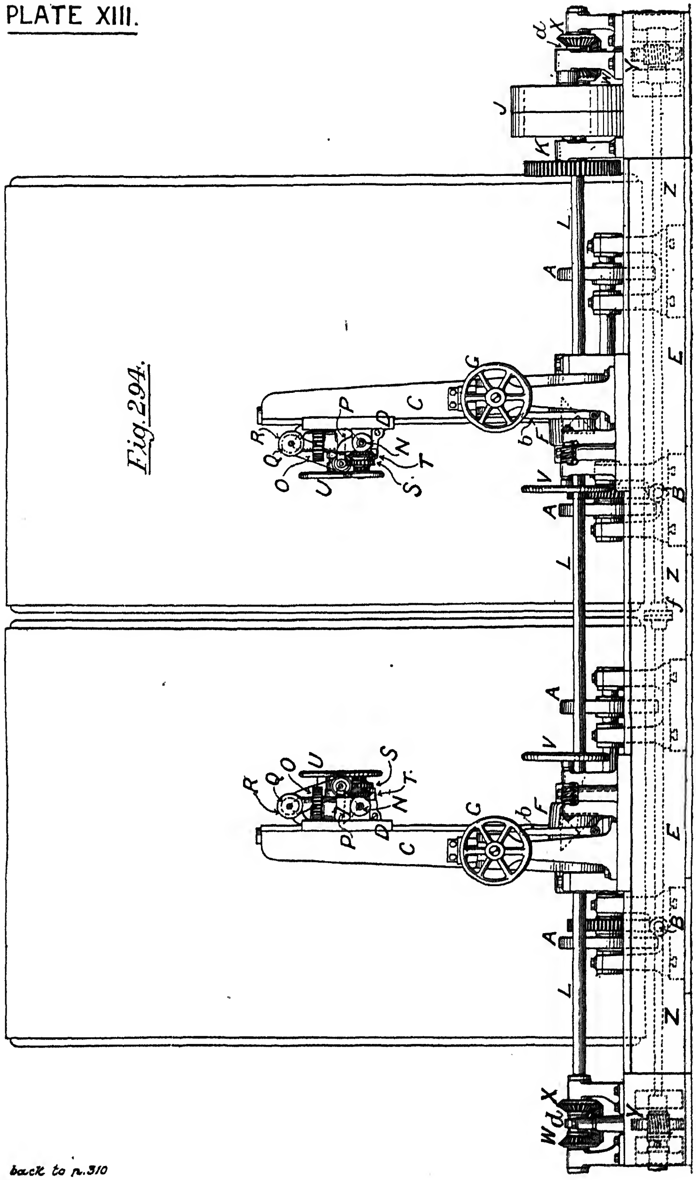


Fig. 294.

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PLATE XIII.

DRILLING MACHINE FOR MARINE BOILER SHELL

(by Hulse & Co.)

SCALE OF FEET.

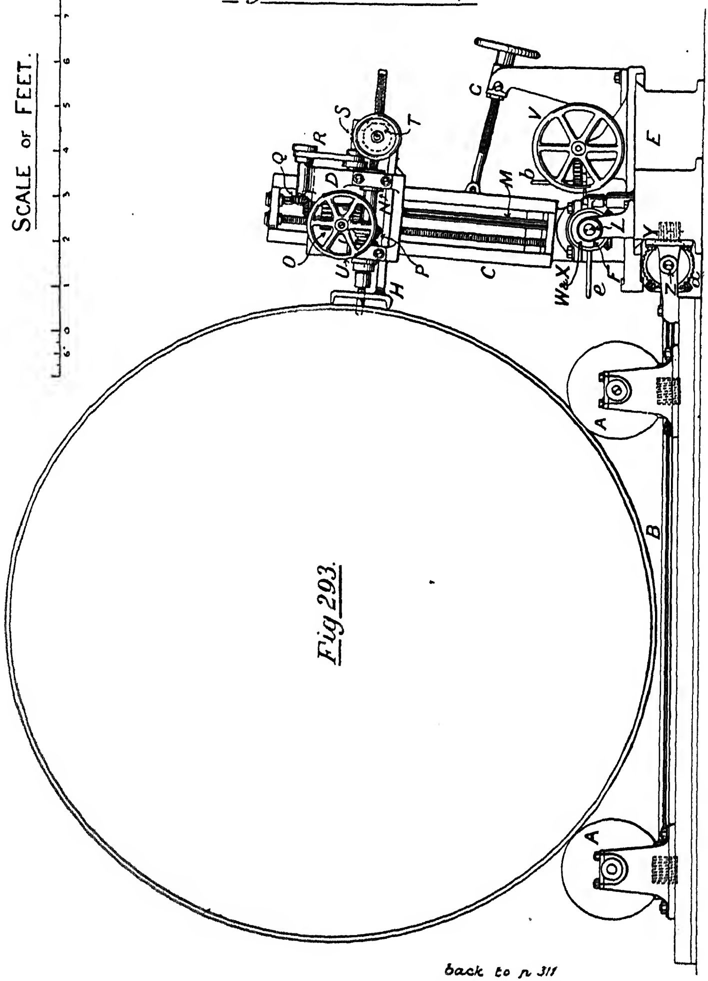
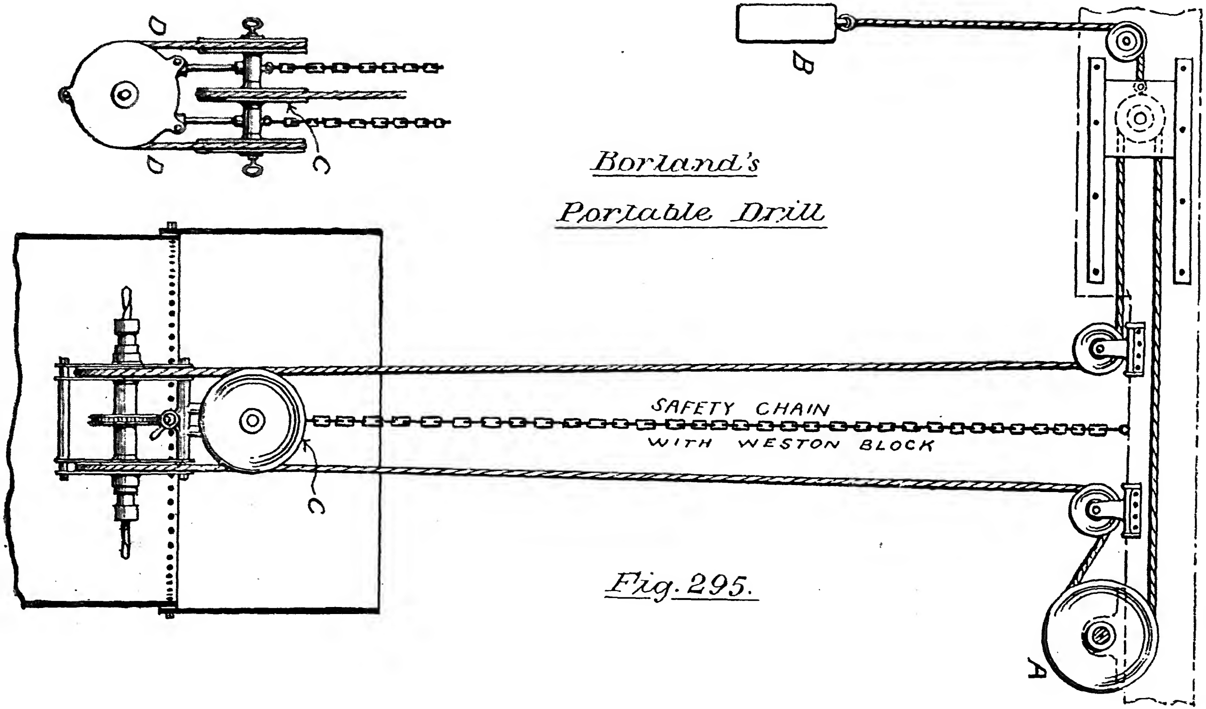


Fig 293.

back to p 311



Borland's Portable Drill

Fig. 295.

proved of great service for various work. It is also [illegible] shown in Figs. [illegible] and [illegible], Plate XIV., [illegible] for some of the Irish railway shops, and made by Messrs. [illegible] & Co. With it [illegible] each drill [illegible] separate [illegible] at [illegible]. If a plate [illegible] operated on, it is laid on the light table [illegible], but if the [illegible] previously built up, then [illegible] is removed, and the [illegible] of the trolley, used instead. The [illegible] either by a hand wheel placed on the square at [illegible], [illegible] the hand [illegible] through bevel gear, or by spokes at [illegible], which rotate a pinion [illegible] within a rack [illegible], one method or the other being [illegible] convenient. [illegible] is the cross slide, raised or lowered by the screws [illegible], coupled by horizontal shaft [illegible], and the drill brackets [illegible] are moveable along the slide by the application of a 'pinch bar' to the [illegible] at [illegible]. [illegible] is the driving shaft and [illegible] the feed shaft. They are connected by the feed gear, [illegible], worms and wheels at [illegible] and [illegible], and the feed shaft is provided with levers [illegible], attached to the drill spindles by links at [illegible]. The drill spindles [illegible] are driven from [illegible] by bevel gear [illegible], and the springing of the two shafts [illegible] and [illegible] is prevented by the brackets [illegible], which support them both. The bevel wheels on the spindles have clutches [illegible], actuated by the balance handles [illegible], so that any or all of the drills may be put in gear at will. The balance weights [illegible] are attached to the levers [illegible], to relieve the weight of the drill spindles, and the set screws [illegible] are tightened against the [illegible] to fix the centres of the drills after adjustment. If only some of the drills be required, clutches [illegible] are disconnected and the drills withdrawn, feed gear being stopped entirely by releasing the clamping handle [illegible], which unites the feed plate [illegible] with the worm wheel [illegible]. The method of operation, then, is to (1) lay the plate on the table in position, and bring the work under the drill by turning the spokes [illegible]; (2) adjust cross slide [illegible] for height, and drill brackets for centres by pinch bar at [illegible]; (3) fix by set screws [illegible]; (4) start shaft [illegible], and pull down the clutch levers [illegible]; (5) bring drills down to work by handle [illegible], and (6) put feed motion in gear by clamp [illegible].

The drilling being done, unclamp the feed, raise drills by handle [illegible], change the strap to loose pulley, and set to another row

PLATE XIV.

MULTIPLE DRILLING MACHINE

(by Hulse & Co.)

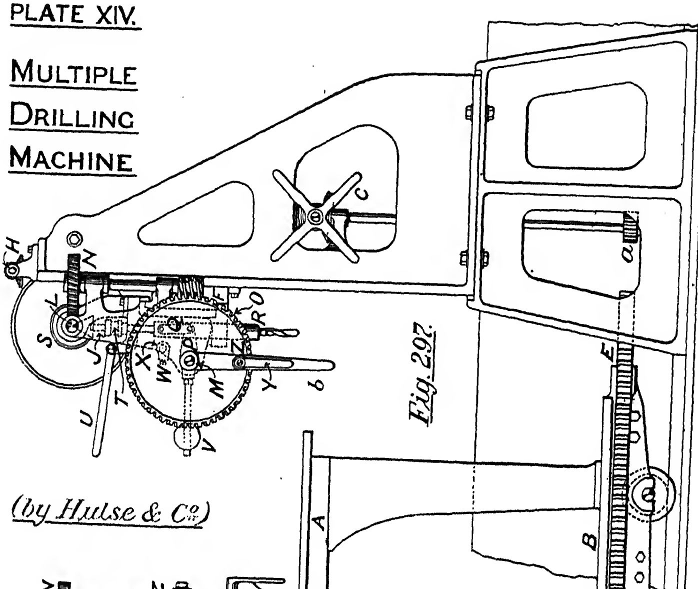


Fig. 297.

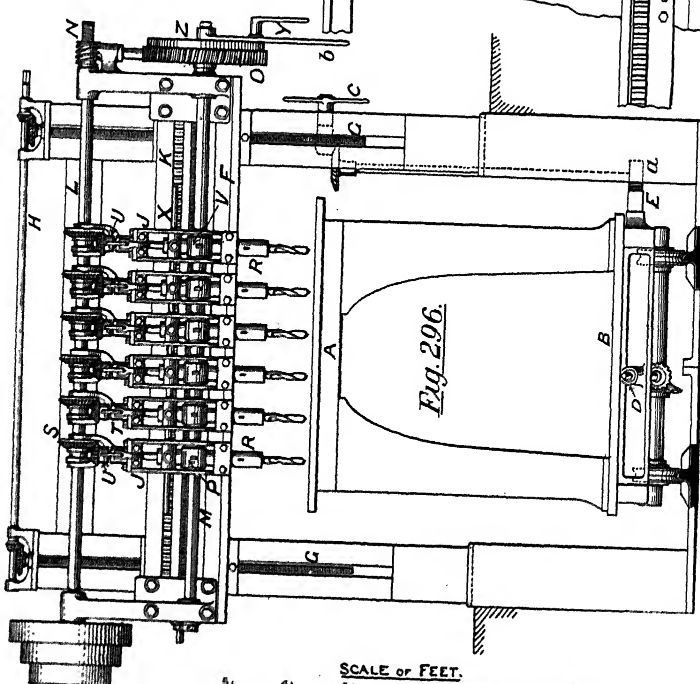


Fig. 296.

SCALE OF FEET.

of holes. Considerable economy results from the application of this machine, which is very well designed in Plate XIV.

Summing up, the great desiderata in boiler drills are rapidity of adjustment and withdrawal of tool, and where possible the introduction of multiple drilling.

Hydraulic Riveting Machines.—It is to the late Mr. Ralph H. Tweddell that the honour of introducing hydraulic riveting belongs. No other method is now used, excepting pneumatic and electric contrivances, which are now being more employed: but steam riveting is entirely obsolete. The advantage of hydraulics for riveting is very great: it is a power that can be conveyed to great distances without appreciable loss, it can be stored till wanted, and the steady and known pressure on the rivet-head, coupled with the increase due to absorption of the momentum of the accumulator weight at the moment of closing, is just the action most desired. (*See Appendix I., p.* 754.)

Large Fixed Riveter.—This machine, on Tweddell's system, is shewn in Fig. 298. The standards A and B are securely connected by two bolts at C, and well designed to resist the stresses caused in closing. A supports the cylinders, while B serves as 'dolly,' carrying the tail cup M, and presenting a nearly flush top surface, for the purpose of getting into corners. The riveting cylinder Y, carrying the heading cup, rides upon a fixed ram T, and within Y is placed the ram U, which advances the annular plate-closing tool V. The auxiliary ram X, of piston form, receives pressure on either face for advance or return: and the tank D, placed 20 feet above the top of the machine, supplies the cylinders T and U with low-pressure water. The pipe E carries this water to cylinder T, and the branch pipe R passes to U, the check-valves Q and S in each case preventing return excepting through the exhaust pipe L. The latter communicates with each of the piston valves, P, O, N, as does the pressure pipe J; P being connected to the back end of the cylinder X, through the pipe *a*: O with the cylinder U through pipe Λ: and N with the cylinder T: while *b* is a constant pressure pipe connecting J and the front end of X. K is a stop valve, and Z an overflow pipe.

We can now understand the action of the machine. The

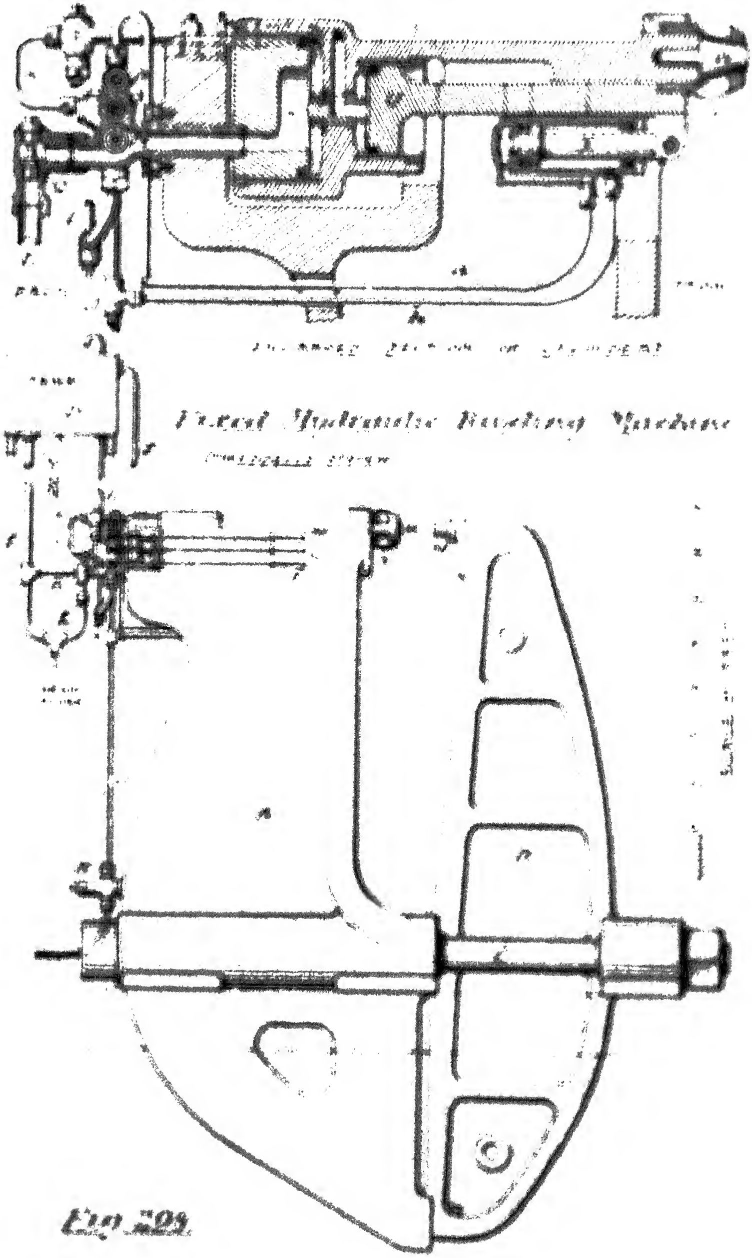


Fig. 229.

boiler seam being placed between W and M, the rivet heated and put in from the side M; lever H opens valve P to pressure, and a right-hand advance is given to the ram X, due to the difference of area of its faces. This pressure, assisted by the head of water passing from the tank, through the check-valves Q and S, carries forward parts U and Y. When W and V reach the rivet and plate respectively, lever G admits pressure water at O through pipe A, to advance the ram U, thus pressing the plates firmly together between tools V and M. And now valve N is opened by lever F, and pressure given to T in turn, thus bringing forward the cylinder Y and the cupping tool W to close the rivet, the pressure obtained being due to the difference of areas of the rams U and T, part of the water from U passing into T through pipe J. The pressure should be kept on the rivet until it cools somewhat, to secure a tight joint, and the three levers are then moved to exhaust, when the pressure *b* pushes back ram X, bringing U and Y to normal position, and lifting the water up L into the tank.

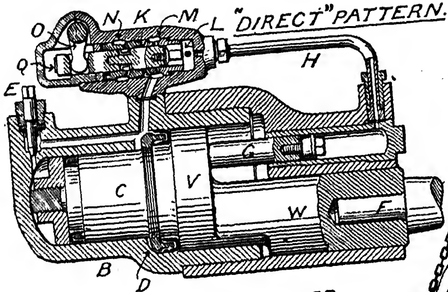
Fig. 298 shews all the later improvements introduced: the plate closing (in 1880) and the use of low pressure water to fill the cylinders (in 1890). The latter is very interesting, and greatly economises high pressure water, which is only used as a film on the back of the tank water, as it were, the fluid being practically incompressible. The plate-closing apparatus prevents 'collars' being formed on the rivet between the plates. In a 100-ton riveter, 60 tons are applied for cupping, while the remaining 40 tons hold the plates together, but ultimately the whole 100 tons is applied to the rivet-head and plates.

Portable Hydraulic Riveters.—Although Mr. Tweddell introduced hydraulic riveting in 1865, his invention of the portable machine did not occur till 1871, from which date Messrs. Fielding and Platt, who then took up its manufacture, were associated with him in the design of nearly all his later hydraulic machine tools. There are two forms of the portable machine known as the 'Direct Acting' and 'Lever' types respectively; their present construction being shewn in Figs. 299 and 300. Referring to the former, frame A is a rigid casting, supporting a cylinder B with direct-acting ram C. There are three

diameters on the ram; C and V to obtain two powers, while W acts simply as guide for the cupping tool F. When the smaller power is required, water pressure is admitted to the annular area D, but if plug E be unscrewed it acts also on the back of C, the pressure then being due to both areas C and D. K is the valve box, containing the piston valve Q, capable, by means of the passages within it, of connecting the annular chambers N and M, or of opening M to L, where the pressure-water enters. G is the returning ram, upon which a constant pressure is exerted through pipe H, and space N communicates with the exhaust pipe J. The handle P acts on the valve lever O, so that if the latter be moved to the left, space M is uncovered and pressure-water enters cylinder B; but if O be moved to the right, spaces N and M are connected, and the cylinder water passes out to the exhaust pipe. The machine is slung by chains R R from a pulley T, provided with worm gear; by turning which from the hanging chain T, the frame may be set at various angles to the vertical within the plane of the paper. Studs also are fixed on the frame at the centre of gravity of the whole, on which are placed the slinging pieces X X, and the worm gear at S turns the frame in a plane at right angles to the previous movement: universal adjustment being obtained by the combination of the two motions. The space between the cupping tools may be adjusted by the insertion of longer or shorter dies, or by packing collars; and the method of riveting needs no further description.

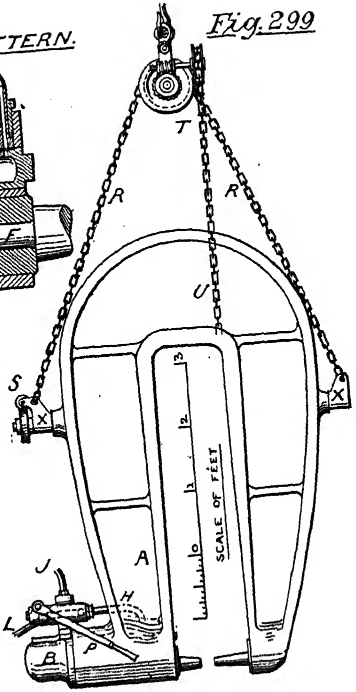
Taking now the lever machine at Fig. 300; A and B are the levers, the first carrying the piston E and the second the cylinder D, while both are connected by the pin or fulcrum C. To avoid another joint the curved cylinder was devised by Mr. Fielding, as well as special tools for its perfect machining: two enlarged sections of it are shewn. The pressure pipe is coupled at J, where a sheath attached to the union preserves the pipe from injury by sudden bending, and the movements of the machine are not interfered with, for the water passes through a swivel joint at K, through the coiled pipe M and the swivel N, then through the pin at N and the arm Q to the fulcrum pin; another swivel R and a short pipe T connecting C with the valve-box. U is the exhaust pipe, led away as required, and the piston valve H is

"DIRECT" PATTERN.



SECTION OF CYLINDER

Fig. 299

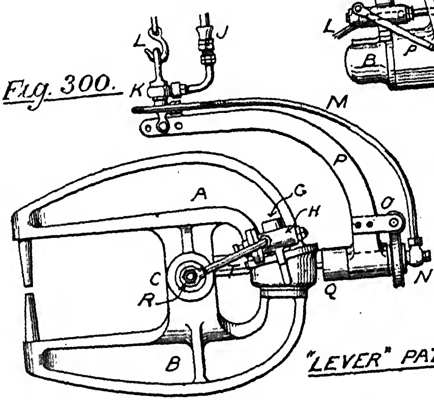


Portable Hydraulic Riveting Machines.

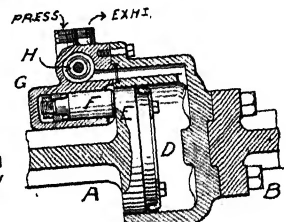
(TWEDDELL'S SYSTEM)

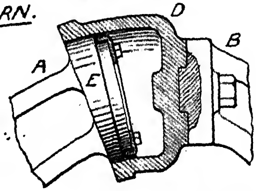
by Fielding & Platt.

Fig. 300.

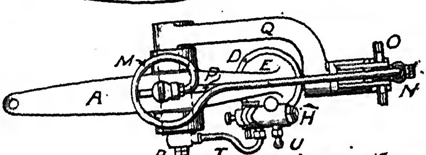


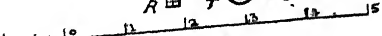
"LEVER" PATTERN.





SECTIONS OF CYLINDER





SCALE OF FEET.

moved by a cam (not shewn) to open the cylinder to pressure or exhaust, while a constant pressure passage [illegible] with cylinder [illegible] so that the constant [illegible] whenever it is opened to exhaust. The [illegible] of gravity of the whole apparatus, and the rivets are [illegible] directly by the [illegible], and [illegible] the [illegible] was that they may be swivelled upon centres [illegible] perfect adjustment to suit the work. The [illegible] gear [illegible] the position round the axis [illegible], and the drawing shews the latest method of hanging the riveter.

The choice of one or other of the machines described depends upon the nature of the work. The direct-acting machine has the advantage of rapidity, but the lever machine can be applied more easily, and reaches more rivets, being therefore useful where the character of the work is constantly changing, and the rivets less accessible.

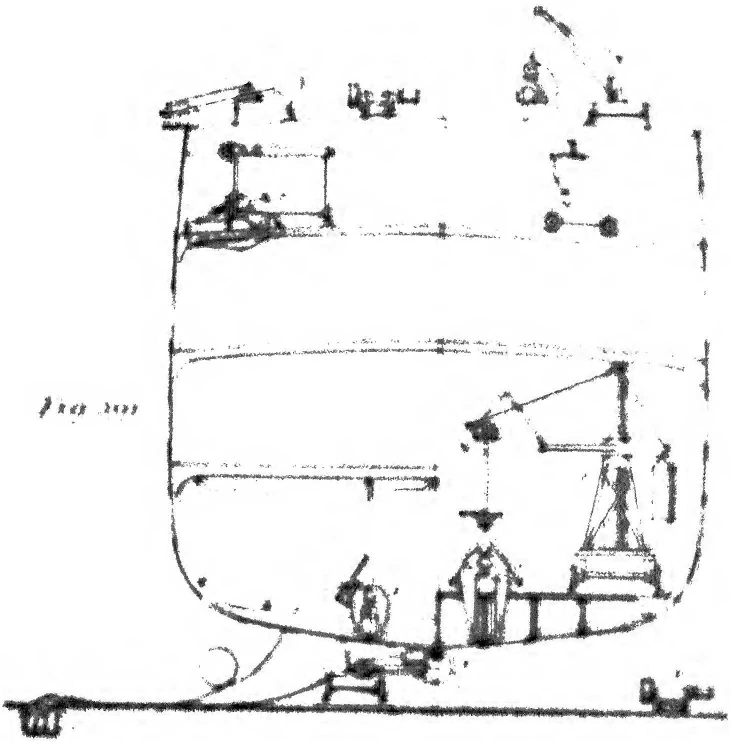
We may now direct the student to Plate XX., which shews Tweddell's system of Hydraulic Machine Tools applied in a Locomotive Boiler Shop. A is a Fixed Riveter, similar to that in Fig. 298, but without plate closer. The handles at c work the crane D, which lifts the boiler, E being the lifting cylinder, F the slewing cylinders, and G the travelling cylinder, each supplied with multiplying gear. H is a smaller crane, where the job is lifted direct from the cylinder. K is a crane for portable riveters, the trolley J having a ram for vertical adjustment of riveter, the horizontal position being obtained by hand. The pressure pipe on K is jointed for horizontal movement of J, and the pipe of J is coiled to give spring during vertical movement. Of the portable riveters, L and M are of the 'bear' type, the former having one and the latter two supporting arms. N and O are ingenious applications of this type, and P is an example of the 'lever' form with very long levers. The small 'bears' at Q, R, and S has been devised for fireholes and foundation rings, being converted from two arms, and the toggle gear at S adapts the fixed riveter to firebox crowns. T is a Forging Press for stamping purposes, and U the 'Pedestal' Flanging Press, detailed at Fig. 289. Crane V is used with this press, and the Travelling Crane W covers the centre aisle of the shop. The striking difference of the cranes

required is very apparent, the importance of ample provision for lifting being a point upon which Mr. Tweddell always insisted.

Plate XVI. represents the interior of a Marine Boiler Shop. B is a Stationary Riveter, exactly as in Fig. 298, and a circular pit C admits a large marine boiler when riveting. As it is difficult to obtain nicety of vertical movement in the travelling crane D, an intermediate cylinder or Hydraulic Adjuster, E, forms a very useful adjunct. The Progressive Flanging Machine F was shewn at Fig. 290, and the crane A lifts the plate to or from the fire. A plan view of the latter is given at G, where the dotted lines shew the plate being heated. H is the Locomotive type of Marine Boiler, much used for the smaller boats, the riveting of which is performed as in Plate XV. A Marine Boiler is given at J, having the furnace mouth riveted round with a small bear K, which also joins the 'Adamson' flues at L. At M the boiler is being closed by a powerful bear-type machine, having plate-gripping tool, and hung from the travelling crane through the medium of the adjuster N. The last-finished flange is here turned outward, as advocated by Mr. Tweddell, to secure good machine riveting throughout; but as many makers prefer an internal flange, to save cargo space or reduce weight, the riveter at P has been recently devised. It is slung from its centre of gravity, and the free arm lowered into the boiler, as shewn dotted. When raised, the latter serves as 'dolly,' and can be adjusted in length to suit various diameters of boilers. A hole must be left at Q, to be covered afterwards by the plate carrying the central nest of tubes, the final riveting of which is performed by hand.

The diagram in Fig. 301 shews the arrangement of hydraulic tools on Tweddell's system applied to Shipbuilding. A is a keel riveter, supported by parallel motion and balance weight, so that it may be raised or lowered to reach the keel in any position, yet remain with jaws vertical. The gunwale riveter at E is similar in construction. H shews a small travelling jib crane, carrying a direct-acting machine for riveting the combings of hatchways. J and K are hydraulic winches, and G a punching or shearing machine. D is another jib traveller carrying the large lever riveter C for finishing the double bottom, and the machine B, supported by a crane with two movements, is for riveting the keelson. A special carriage F

Fig. [illegible]

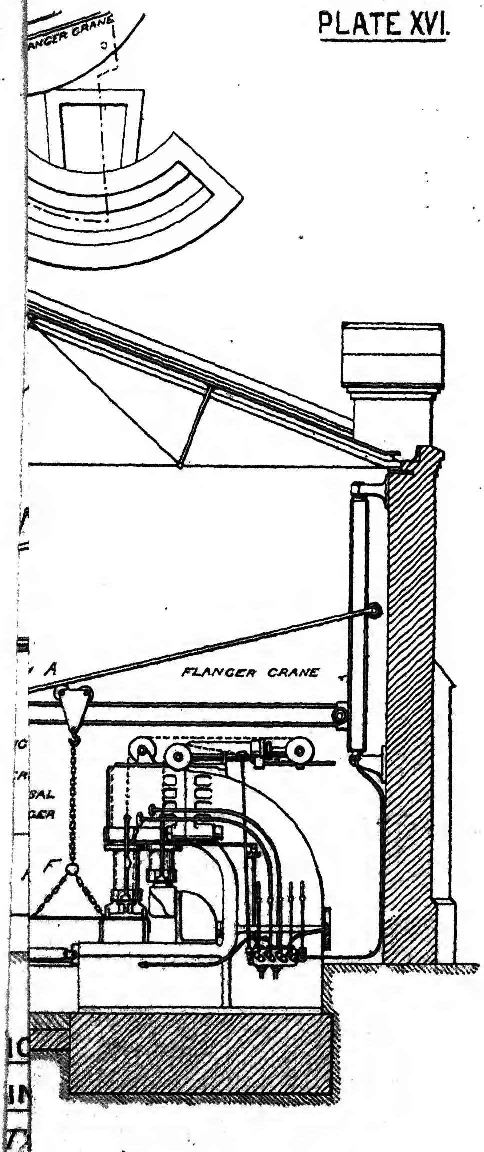


Section of a Ship in course of construction showing Tweddell's Machinery in operation.

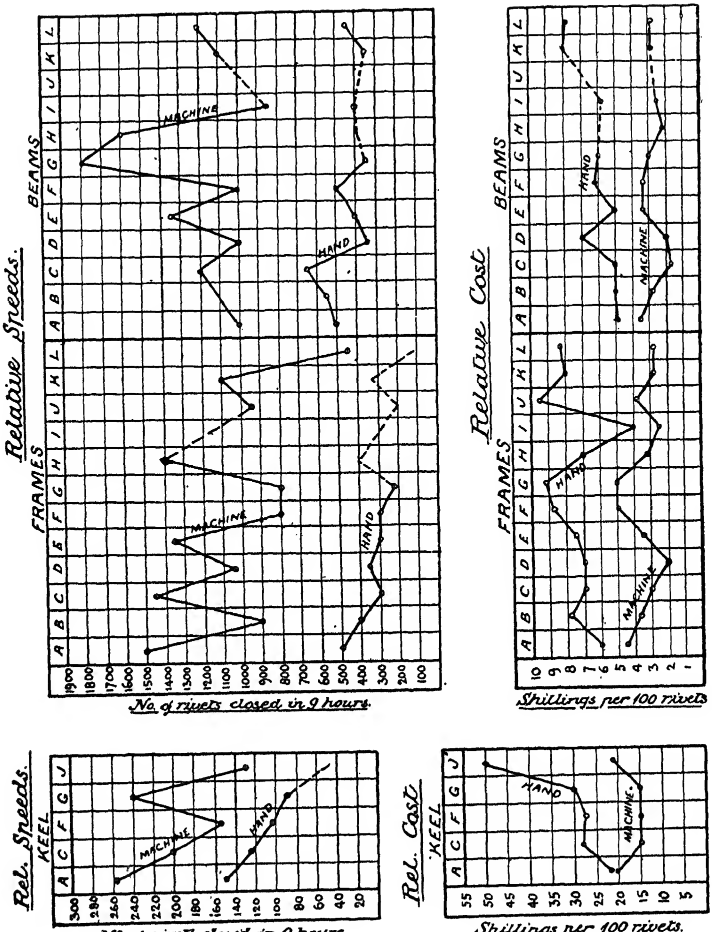
would pages of description. Indeed, the Forth Bridge, all the great bridges of India, and the Tower Bridge, could not have been erected up without these wonderful machines.

The pressure used in the hydraulic mains for boiler shops, &c., is usually 1500 lbs. per square inch, but is sometimes exceeded, being often [illegible] lbs. (See *Appendix II.*, § [illegible].)

PLATE XVI.



face p. 320.



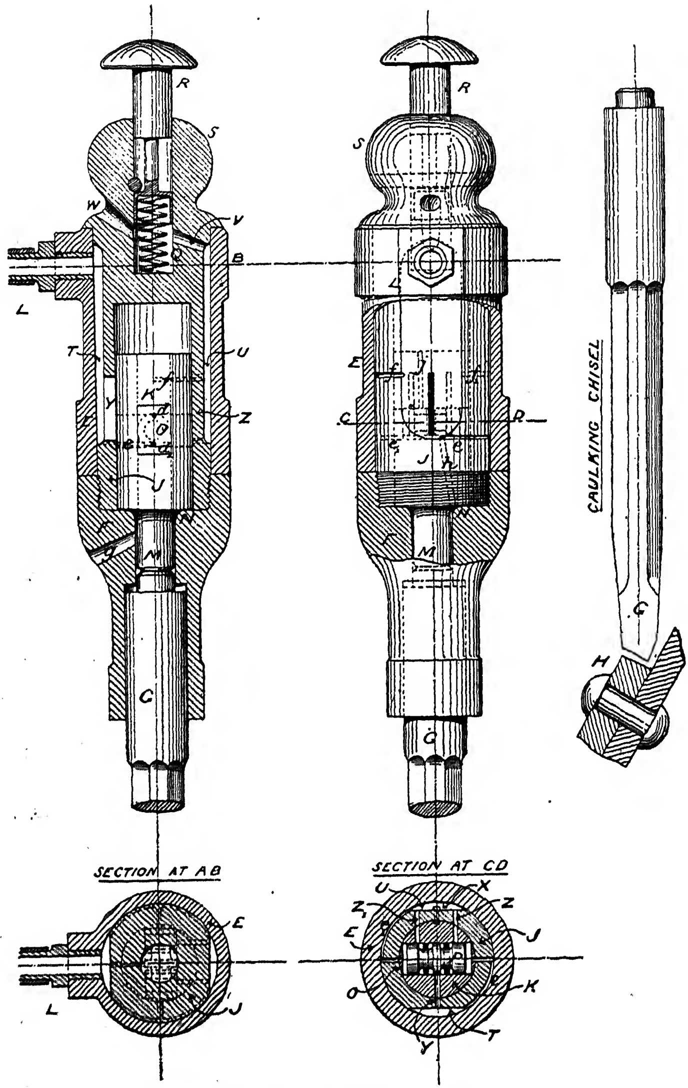
Comparative Speed & Cost of
Hand & Machine Riveting (TWEDDELL)

Letters A, B, C, D &c. indicate the results
from various shipyards.

Fig. 301a.

√ **Pneumatic Caulker.**—This, an American invention, was first introduced in 1890. As shewn in Fig. 302, it was formerly made by Messrs. Crossley Bros., and would do the work of three or four men. E is a jacket held in position by the cylinder J, screwed into the nose-piece F. The caulking chisel G is loose, but placed within F when required. The piston contains a piston-valve P, vibrating at right angles to the piston's axis, the slide hole being closed by slips O O, dovetailed into K. The starting valve R, when in the position shewn, allows the compressed air, after entering at L through a strong indiarubber tube, to pass through the piston by T and U, then harmlessly out by the passages V and W; but if R be pressed down the passages V W are closed and the machine operates in the manner to be described. Key X allows the piston to slide vertically, but prevents axial rotation. Y is a passage from T to the piston, and T and U being formed by flats in S, are not in communication with each other. There are two passages from the piston to U, seen in plan at z z_1, while in the piston itself one passage j communicates with the top of the cylinder and another h with the bottom. In addition, two holes d d_1 are made in the slips O O, and grooves e e_1, f f_1 are in connection with these holes at certain times. One other point must be noticed—the hole g is the exhaust outlet when in working order, but M fits the hole in the nose-piece so that air cannot escape when the piston is at the bottom of its stroke. If, however, K be lifted to the top position, M will be found just of a length to disclose an annular space round the curvature N, and the air is free to pass out at g.

Having noted all the parts, we can now describe the working of the tool. The workman, after placing the chisel G in the nose-piece, holds the former with his left hand against the seam of the boiler as at H, while with his right hand he grasps the boss S, pressing the head R upon it, thus practically closing the passage U. The air passes through T and Y, but cannot get further. Hole d_1 is now in communication with passage e, however, so the air enters the valve chamber from the right and moves P to the left. This allows the pressure to act through h on the bottom of the piston, and the up stroke is made. While this air exhausts at g, the hole d, being now in communication with f, the valve is



Pneumatic Caulking Tool

Fig. 302.

SCALE OF INCHES

[illegible]

Tube Expanders. [illegible] to withstand the rough treatment to which they are subjected. The earliest metal employed was copper, but that was expensive. Brass is now extensively used for the smaller sizes, and [illegible] wrought iron for the larger tubes. [illegible] has been much used, and appears to possess [illegible] properties, being intermediate between the brass and copper. [illegible] as a test of fitness the material must allow of being [illegible], and having the ends enlarged [illegible] capable of [illegible] of the original diameter. The treatment which tubes receive in the boiler shop consists in their expansion to fit tightly in the tube plate holes, and beading at the fire-box end. [illegible] large tubes are often lap-welded, smaller ones are [illegible] on a pointed mandrel [illegible] Fig. [illegible], the [illegible] being long enough for the whole tube. The process reduces the ductility of the material, so the tube ends must afterwards be annealed, while the rest of the tubes should be perfectly straight and uniformly elastic.

A Tube Expander is shown in Fig. [illegible] is the [illegible] tube plate and tube, [illegible] the body of the tool, [illegible] three rollers [illegible] in between [illegible] and the collar [illegible] an adjustable gauge [illegible] the position of the rollers with respect to the tube plate, and [illegible] a taper mandrel provided with a tommy bar [illegible]. The [illegible] the rollers are rotated by rod [illegible], which is at the same time pressed forward till the tube is well expanded to fit the tube hole. [illegible]

Manufacture of Tubes.

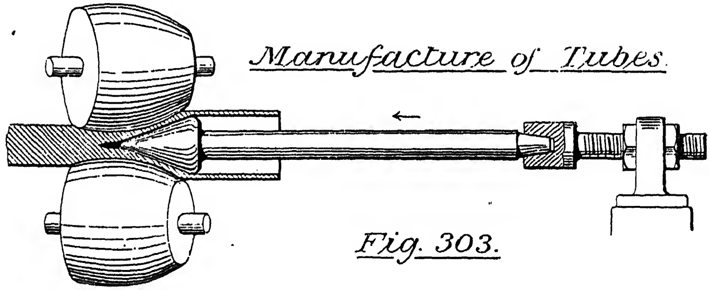


Fig. 303.

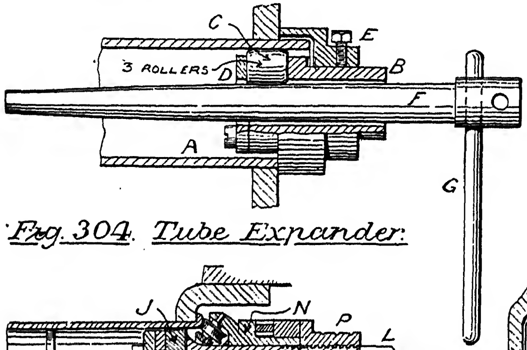
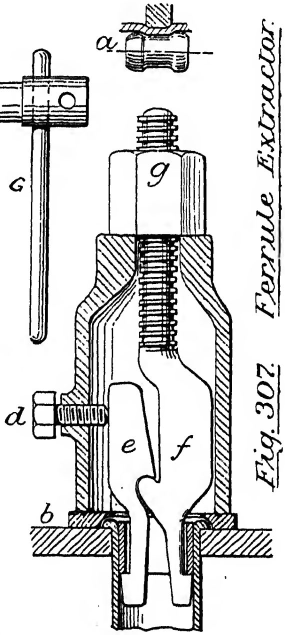


Fig. 304. Tube Expander.



Ferrule Extractor.

Fig. 307.

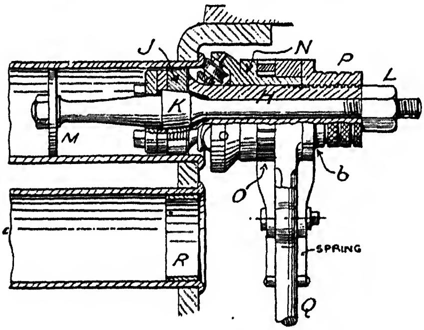


Fig. 305. Tube Beader.

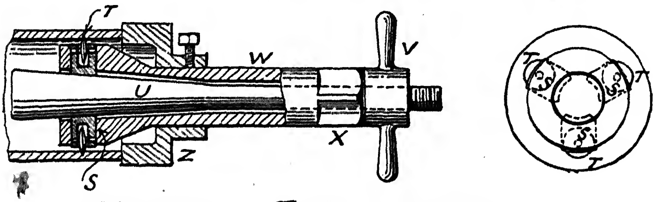


Fig. 306. Tube Cutter.

Yarrow recommends the roller shewn at *a*, and advises that the hole in the tube plate should be rimered to the same taper as the mandrel F.

Tube Beader (Fig. 305).—H is the body of the tool, containing three jaws at J capable of sliding radially, and moved outward by the taper part of bolt K when nut L is tightened up. This is done when the beader is put in place, the disc M serving as steadiment. The collar N holds three rollers, placed at such an angle as to do the work efficiently, and a ratchet wheel O is keyed to N. P is the feed nut, and the ratchet arm Q rides loosely on N, the latter being driven by Q, like the drill in Fig. 215. But there is one depression *b* in the rim of the feed nut P, so that when Q has, by its vibrations, brought N round by one revolution, the feed nut is automatically advanced by a small amount. The firebox ends of the tubes being excessively strained by the great variations in temperature there occurring, beading protects the joint, while the ferrule R, in addition, secures the rigidity not obtainable by simple expanding.

Tube Cutter.—As it is impossible to gauge the length of the tubes accurately beforehand, the tool at Fig. 306 becomes necessary. Three bearings S S S, capable of radial sliding, support hard steel discs T T T, which are the cutters. The tapered bolt U advances these bearings outwardly when tightened up by the nut V; this may be termed the feed. The tool body W has a square at X and an adjustable gauge at Z, by which the cutters are set. The gauge being fixed, the tool inserted, and nut V screwed up, a spanner on X rotates the whole. Then V is tightened, the operation repeated, and so on till the tube is cut through.

Ferrule Extractor.—As tubes have to be withdrawn and replaced, and the ferrule is the most troublesome portion to remove, the extractor at Fig. 307 has been contrived to meet this difficulty. The washer *b* is first placed against the tube plate; then the set screw *d* released to allow the jaws *e f* to enter. When all are in position the screw *d* is advanced to press the jaws against the tube, and the nut *g* then tightened with a long spanner and the ferrule drawn out. All the four foregoing tools are supplied by Messrs. Selig, Sonnenthal & Co.

Electric Welding.—This important process, first introduced in 1885, has proved of great advantage in satisfactorily uniting pieces unattachable by ordinary means. Among these articles are boiler plates, which must be our apology for introducing the subject here. Wrought Iron, or in a less degree Mild Steel, were the only materials previously weldable, and even then the joint had but 70 per cent. of the strength of the solid material —a serious matter with crane chains, where every link is welded. Scale might form between the weld, the heating could not be seen openly, and might neither be even nor thorough; objections all absent in electric welding.

Electric energy consists of two factors—electromotive force (or pressure) multiplied by the current (volts × ampères). If this energy pass through a *good* conductor, nothing is observable in the latter; if a *bad* conductor be presented, the current will not pass; but an *indifferent* conductor will allow some of the energy to pass, while the rest is converted into heat on account of the resistance, the amount of heat energy produced being equivalent to the electric energy destroyed. The metals we most desire to weld are in the class of semi-conductors, and there is no difficulty in raising their temperature to welding point by the electric arc; but the heating effect of a current is independent of the pressure or potential, *depending only on the value of the current*, and it follows that the energy from the dynamo must be *transformed*, so as to obtain a low *voltage* with a high *ampèrage*. Every one knows the galvanic battery and induction coil, where a current of low potential becomes one of high potential after passing the coil, though at a sacrifice of current value, the energy remaining the same. Transformers serve the same purpose, being similarly designed, and it depends which side of the transformer we are on as to what ampèrage we obtain.

There are two processes employed in electric welding, the 'Thomson' and the 'Benardos,' named after Professor Elihu Thomson and M. Von Benardos respectively. The first consists in using the pieces to be united as the poles, and the second in using one of the pieces as the negative pole, while the positive pole is supplied by a rod of carbon, held in the hand in the manner of a soldering bit. The electric energy is obtainable in

either case by one of two methods—(1) from an 'alternating' dynamo, the 'current' being increased by passing through a transformer; (2) from storage or *secondary* batteries, which take their energy from continuous dynamos. The welding apparatus is not thereby altered. A general diagram in Fig. 308 shews the direct method combined with the Thomson process, where A is the dynamo, B the transformer, and C the welding apparatus. Two wires are clamped in position at D, and end pressure put on by the screws, the current switched on at E and regulated at F. The ends of the wires are previously brightened, and a flux of powdered borax interposed. After welding, the bar or wire is removed and hammered to size.

Energy remaining the same, the following examples will shew the variation in ratio of potential and current for various purposes:—

1. For arc lighting:	2500	volts at	10	ampères.
2. For incandescent lighting:	100	volts at	250	ampères.
3. For welding:	$\frac{1}{2}$	volt at	50,000	ampères.
4. For welding:	$\frac{1}{4}$	volt at	100,000	ampères.*

No. 3 would weld steel bars $1\frac{1}{2}$ inches in diameter in less than two minutes, while No. 4 would do the same in one minute, absorbing 35 H.P., but only for a short time. The great advantage of electric welding lies in the local character of the heating, which prevents the spoiling of a finished piece of work.

We will now turn to the Benardos process, shewn in Fig. 309. It is there worked by accumulators—the method most preferred. The batteries being charged from a shunt-wound dynamo, they are connected to a switchboard A, so arranged as to throw them out in sets of five. From this board the current passes through resistance coils for further regulation, and then through the welding tool B, the pieces to be welded, and back to the accumulators. Fifty cells are usually employed, and, if two boiler plates of about $\frac{7}{16}$ inch thick are to be united, the tool carries a very

* NOTE.—Only strictly correct in the Thomson process, where energy absorbed is due to true resistance. The Benardos process uses the *arc*, and energy is required to produce light, viz., to volatilise the carbon and render it incandescent: amounting roughly to 30 volts in addition.

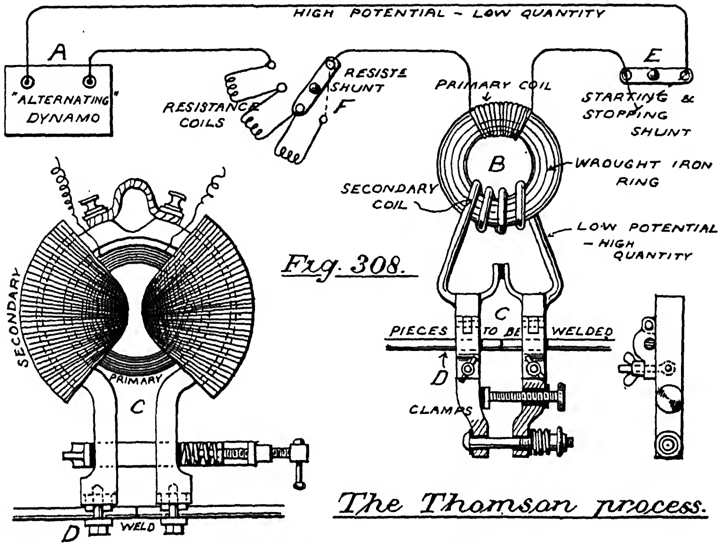
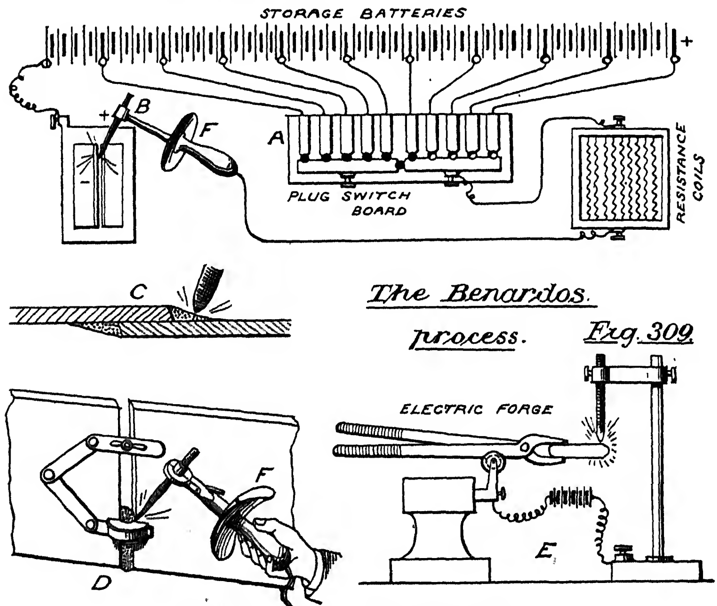


Fig. 308.

The Thomson process.



The Benardos process.

Fig. 309.

Electric Welding.

[illegible] while the boiler [illegible] left hand [illegible] at the weld [illegible] which may be [illegible] pleted by testing the [illegible] handles [illegible] the anvil, and touching [illegible] with the piece of work. The intense heat of the [illegible] while, with [illegible] the [illegible] (see Appendices [illegible]).

Having described the various tools and apparatus used by the Boiler-maker and Plater, we will proceed to examine the kind of work to be performed by these men and the structures which they are to build. We will consider Boiler Work first.

Cornish and Lancashire Boilers.—These are only dissimilar in the number of their internal flues. The Cornish boiler has but one, while the Lancashire boiler is provided with two, so we have considered a drawing of the latter only to be necessary. This is given at Fig. [illegible], where A is a longitudinal section, B and C end views, D is an external view of the front, and E of the back plate. The front plate is flat, being fastened to the first shell plate by an angle ring a; but the back plate is flanged at b. The shell plates are numbered from the front thus: 1st, 2nd, 3rd, &c., and as their dimensions would be about [illegible] feet by [illegible] feet, giving about [illegible] square feet area, they may each be made in one piece, as will be seen on referring to the table of maximum sizes (p. [illegible]), where a ½ inch plate may be [illegible] feet long, [illegible] feet wide, or [illegible] square feet area. We are enabled, therefore, to keep the joint near the top of the boiler, out of the water space, and easily reached after the boiler is set in brickwork. The front and back plates are stayed from the shell by "gusset" stays *d d*, which are oblique plates attached at each end by a pair of angles, and are further supported by the longitudinal stays *e e*. The fire-bars being laid in the flues at *f f* on supports, the heated gases pass

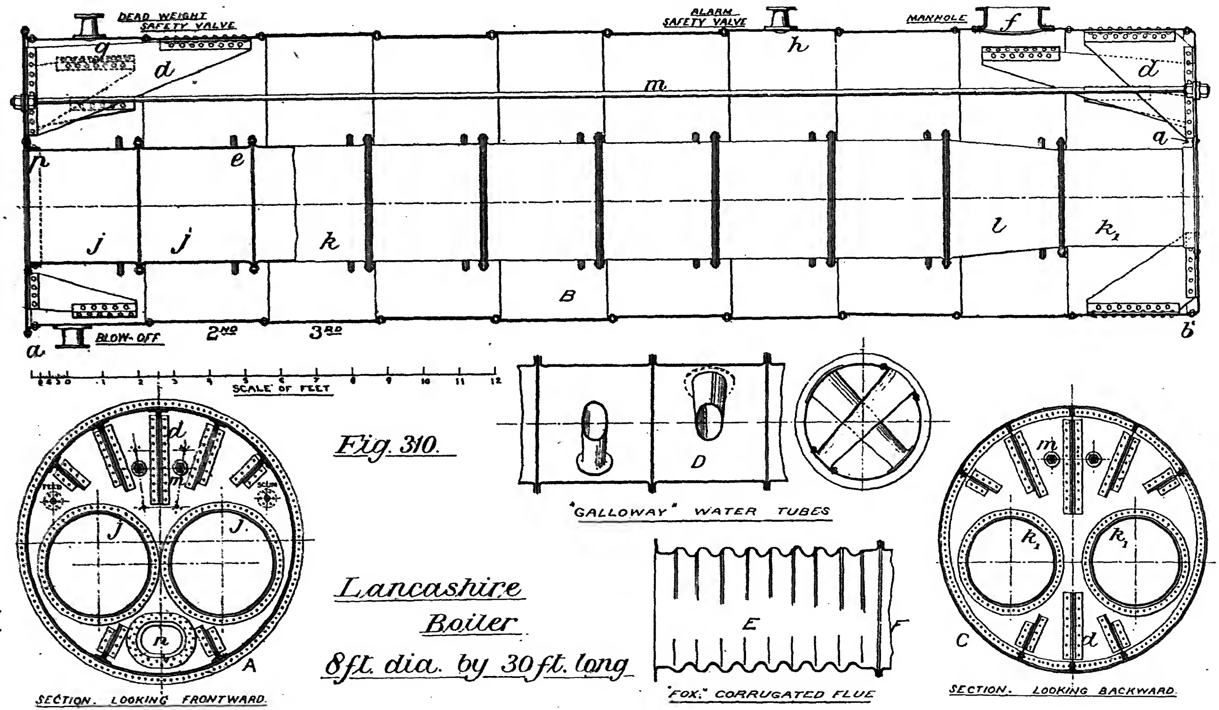


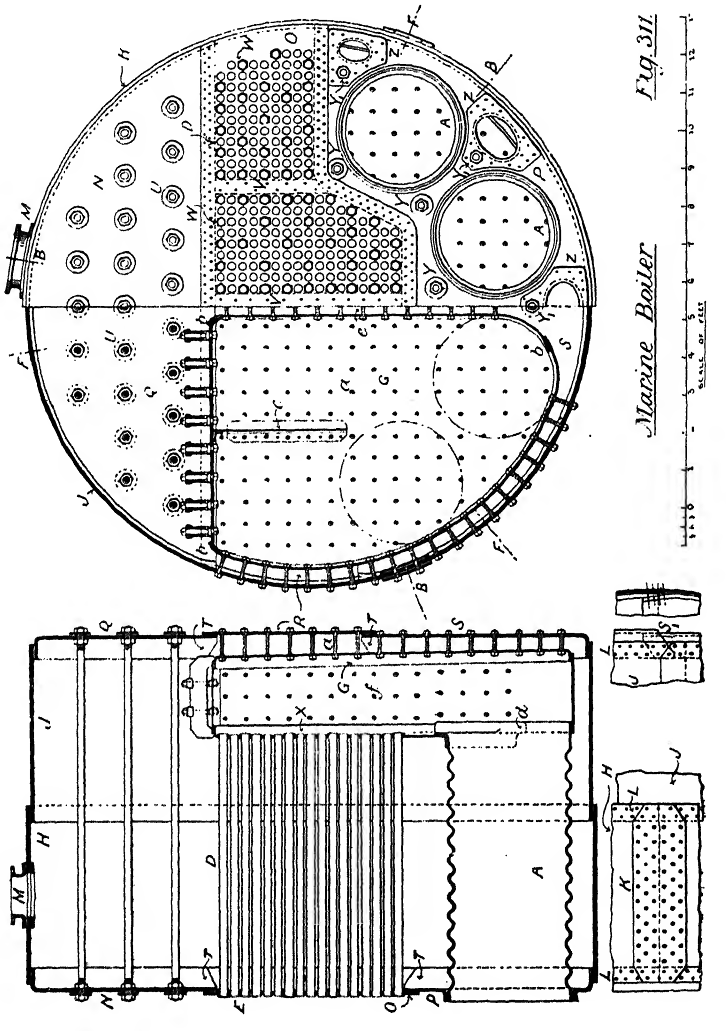
Fig. 310.

Lancashire Boiler 8ft. dia. by 30ft. long

through to the end k_1, thence by brickwork flues, along the bottom of the boiler to the front, again to the back end by the brick side flues, and away to the chimney. The internal flues are therefore at a greater heat than the rest of the boiler; this, producing expansion, necessitates the introduction of elastic portions. The flues, moreover, are in danger from collapse, for a cylinder, although strong when pressed from within, is unstable when pressed from without; so strengthening rings are applied at various distances along the circumference. But as joints have to be formed, on account of the great length of the flues, it is customary to make provision for elasticity lengthwise, and rigidity of cross section also, at these places, the most usual method being by the introduction of the Adamson flanged seam at *e*. This joint has the advantage over other methods, of shielding the rivet heads from flame, and a slightly projecting annular strip is placed between the flanges for caulking purposes. The space between the tubes being small, the seams are made to 'break joint' longitudinally, so as to be easily got at when necessary. Conical 'Galloway' water tubes are sometimes inserted, as at D, for intercepting the heat more satisfactorily, the smaller end being passed in at the larger hole. The flues are joined to the end plates by angle rings, and their diameters decrease at k_1, the connection being formed by the conical portion *l*. The manhole edge at *f* is strengthened by a riveted ring, always added when a large hole is removed; and the mudhole *n* is similarly treated, a portion of plate being left all round, on which to place the internal door. Holes are cut for various fittings, as at *a*, *g*, and *h*. The circular seams are single riveted, but double riveting is used for the longitudinal joints, because any boiler receives but half the stress longitudinally that it does in a circumferential direction (*see p.* 398).

Fox's corrugated flues, shewn in section at E, are extensively used for the furnaces of many boilers, taking the place of the two pieces *j j*; while F is equivalent to the portion *k*. The corrugations give not only strength and elasticity, but a larger heating surface.

The proportion of length to breadth in the boiler shewn is the largest allowed; more often the length is about two-thirds of that given. (*See Appendix II.*, *p.* 827.)

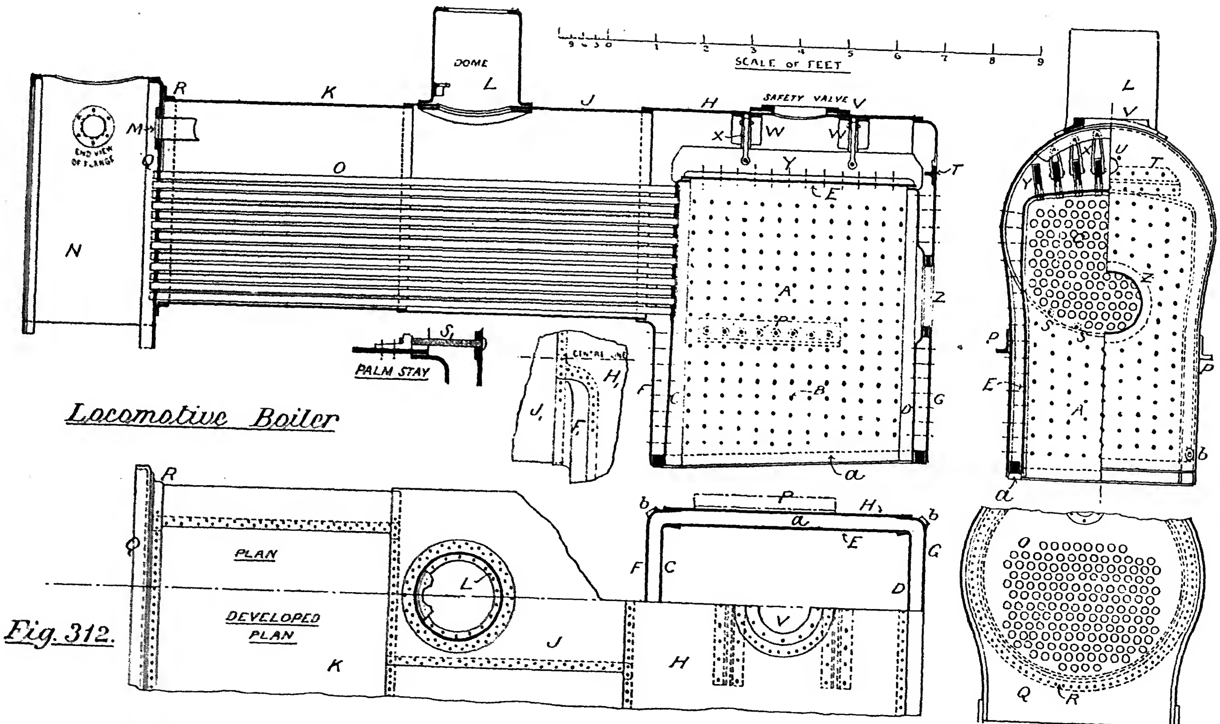


The Marine Boiler, as at present constructed, is shewn by the two views in Fig. 311. The number of furnaces depends on the size of the boiler; in this, a large example, there are four,

A A. The combustion chambers are also variously divided, there being from one to four per boiler; two are shewn at G G, each having a platê C, to split and assist the draught. The heated gases, rising from the fire at A, pass through the combustion chamber G and tubes D, to the uptake, which is placed at E to cover the tubes. The boiler is cylindrical, but with large flat ends which require a good deal of stiffening, for flat portions in all boilers are weak. There are two belts of shell plates $1\frac{1}{4}$ inch thick, the first H and the second J, each being, on account of its large circumference, divided into three, and connected by double butt-straps, with treble riveting, as in plan at K. The division is uniform and is seen on the front elevation; where F F F represent the joints of the first, and B B B those of the second plate. The circumferential seams are double riveted, as at L L, and the man-hole is placed at M, with strengthening ring. Sometimes a separate dome is connected to the top of the second plate, but just as often the steam valve is applied direct to the boiler; in any case the dome is simply a horizontal cylinder with dished ends. The front and back-plates are divided into three, N, O, and P shewing the parts of the front-plate, while Q, R, S are those of the back-plate. N, P, Q and S are each $\frac{7}{8}$ inch thick, but R is only $\frac{3}{4}$ inch, and O is $\frac{13}{16}$ inch. They are all flanged and riveted as shewn, O being cut out a suitable shape to take the nests of tubes, while R is rectangular. Where three plates overlap, the middle thickness is drawn out as shewn at S_1, which is a plan of the joints T T. Longitudinal stays, for the steam space, are supplied by bolts U U, having large washers to distribute the pressure. The plate O is necessarily stiffened by double thickness at the seams, but there are also stiffening plates V V riveted on the inside, and stay tubes W W, shewn by their nuts, support both plate O and combustion chamber tube-plate X. The other tubes, ferruled at the firebox end, and expanded at the uptake end, act also as stays. The plate P is stayed by bolts at Y Y_1, and the manholes are stiffened by riveted plates at Z. The three bolts marked Y_1 pass right through to the back-plate S, which is further strengthened, together with R, by screwed stays at *a*, which are bolts screwed their whole length and fitting into holes tapped in the plates. The combustion chamber back-plate $\frac{1}{2}$ inch thick, shown at G, is a simple flanged

plate; but the tube-plate X, $\frac{13}{16}$ inch thick, is throated to fit the furnace flue. The top and side plates, $\frac{1}{2}$ inch thick, are in three pieces, with joints at *b b b*, and wherever three thicknesses superpose, the mid plate is feathered, as at *b*. Screwed stays $1\frac{1}{4}$ inch diameter, 7 inches apart, are fixed between the chambers at *e* and at the sides, while the roof is supported by girder stays which each consist of two plates resting by their ends on the roof seam. Between these plates are passed collar bolts, which, after being screwed into the roof and fastened by nuts, are tightened against special washers on the girder. The furnace flues are of the Fox pattern, flanged to the throat plate as shewn.

The Locomotive Boiler (Fig. 312) was the earliest form of multitubular boiler, and has served as pattern for many other steam generators. The firebox A is cubical and of $\frac{1}{2}''$ copper-plate, thickened at the tubes to $\frac{13}{16}''$. The back plate D is flanged, and dished round the firebox hole to the form shewn, the tube plate C being also flanged. The top and sides are in one piece E, and all these plates, being flat and weak, are supported from the outer shell by screwed stays riveted over. The latter are $\frac{7}{8}''$ diam. and 4$''$ pitch, and must be of copper, to avoid corrosion by galvanic action, which frequently occurs next the firebox plate. The shell top and sides are in one plate H, cut out as shewn at H_1; the throat plate F is flanged to join the barrel and the firebox shell; and the back plate G is also flanged. The foundation ring *a* serves as a distance or closing piece when fastening the shell to the box, and a similar piece is required at Z, called the firehole ring. Mudhole bosses *b b* are welded on the solid plate, and tapped for tapered screw plugs. A hole is cut in the top of the shell at V for a double safety valve, and the plate stiffened by a wrought-iron valve seating. From angles on the shell roof at W are hung the sling stays X X, supporting the girder stays Y, the latter being solid forgings, and the stay bolts taking the form of tap bolts. T is a stiffening angle for the shell back, and P P are expansion brackets which rest on the engine frame. The firebox tube plate, besides the ordinary screwed stays, has four palm stays at S S, which are seen in detail at S_1. Two plates, K and J, form the boiler barrel, and each makes a complete circle, the joints being shewn in plan, well out of the water space. The dome L



Locomotive Boiler

Fig. 312.

is welded from one piece of plate and flanged over as shown, a stiffening ring being placed round the hole in the boiler, and an angle riveted within the dome to support the regulator pipe. The tube plate G is attached to the boiler shell by angle K, and the smoke box plate wrapped round as shown, but as the latter is not considered part of the boiler, we shall not discuss it further. A hole is cut in the tube plate at M to receive the copper steam pipe, which is expanded to fit, and the tube holes are spaced as at the firebox end, excepting that they are all lifted higher by a small amount to clear the barrel plate. The circumferential seams are all single riveted, and the longitudinal seams double riveted. It will be noticed, in the end views, that the firebox is contracted, because it must fit between the engine frames, but after rising clear of these it may be enlarged.

The Vertical Boiler appears under several shapes. The chief difficulty has been to keep the heated gases in the firebox sufficiently long to allow of their yielding a reasonable amount of their heat to the water. Baffle plates, bent flues, cross water-tubes, have all been used, but the most effective construction seems to be that which imitates as closely as possible a short locomotive boiler. It should be so built that a man can get inside for repair or cleaning. The drawing at Fig. 313 shews Waites & Fraser's patent boiler, a modification of Cochrane's, having the advantages required. A is a dome-shaped firebox, and B a connection to C, the combustion chamber. The shell is a vertical cylinder, made of three tiers of plates, and having a dished roof E stayed by four gussets F F. A manhole G communicates with the combustion chamber for the purpose of getting at the tubes, and the firehole is at H. The tubes J are expanded into the tube plates, and the smoke box K is affixed afterwards. This form of boiler is found very efficient and a considerable improvement on the older types of vertical boiler, which lost much heat.

The Tubulous or Water-tube Boiler differs totally in design from any of the preceding. Fig. 314 shews a sectional elevation of the boiler, with its brickwork setting. A number of comparatively small lap-welded tubes at A are inclined over the fire, and connected at their ends by *zigzag* chambers at B and H,

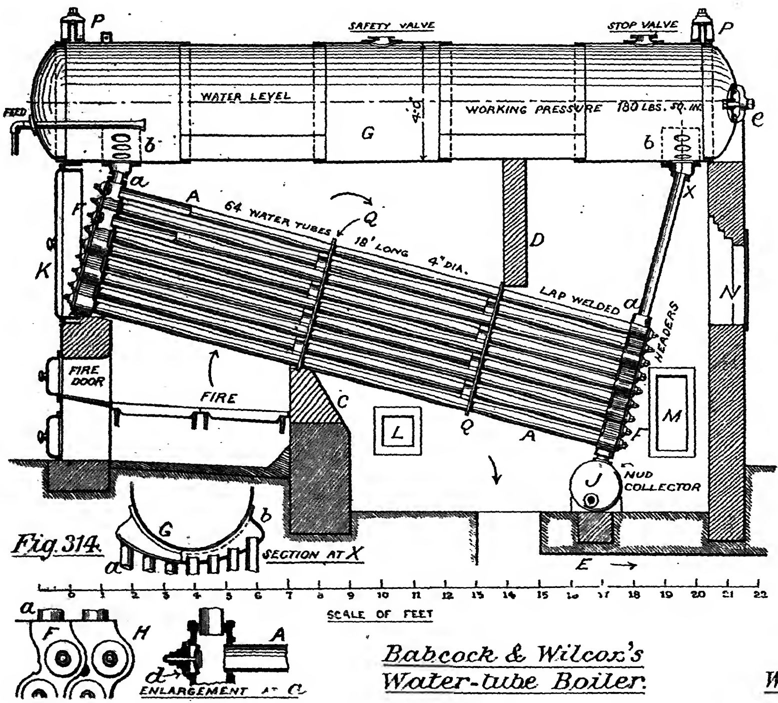


Fig. 314.

Babcock & Wilcox's Water-tube Boiler.

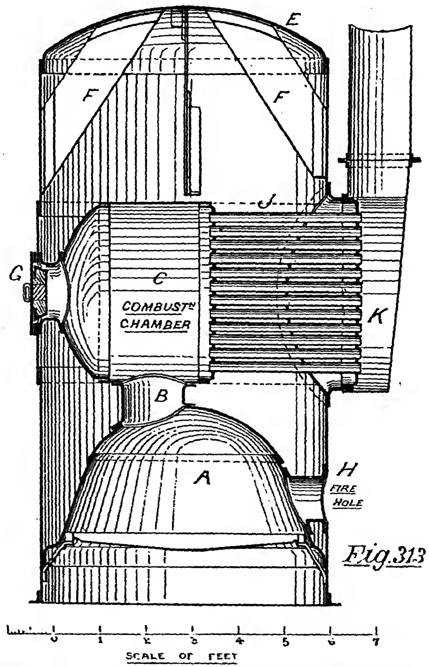


Fig. 313

Wailes & Fraser's Vertical Boiler.

termed headers, which fit closely together. Every header is connected, by a tube *a*, with a collecting chamber *b* at each end of the receiver G; and all the tubes are expanded into their respective sockets, the necessary holes at *d* being closed by covers of 'mudhole' pattern. The water rises to the centre of the receiver, which therefore serves both as dome and part of the boiler. There is a cleaning hole at *e*. J is a cylindrical mud collector, while K, L, M, and N are soot doors; and the draught is compelled to follow the tubes, by reason of the division walls C and D, the flame plates Q Q, and the position of flue E. The receiver is held in place by the girders P P, bolted to the brickwork. The headers are usually of cast iron, though wrought-iron ones have been recently constructed, and plates Q Q, with firebrick distance pieces, serve to stay and support the tubes intermediately. The chambers *b b* are flanged and welded from wrought-iron plate, the tubes are of wrought iron or steel, and the receiver of steel plate. The flue may be at N instead of E.

These boilers have been much favoured recently by electric-lighting engineers, on account of rapid steam-raising properties, and immunity from accidents due to the small diameter of their tubes, with relatively great strength; but they require considerable cleaning and repairing. (*See Appendices, pp.* 828, 993, *and* 1061.)

Geometry required by the Boiler Maker.—This is not of a difficult kind, but involves one or two intersections of solids, and development of the contact line upon either of the solids when their surfaces are laid flat. He must know the relation of circumference to diameter of circle, thus—

$$\text{Circumference} = \text{diameter} \times \pi$$
$$\text{and } \pi = 3{\cdot}1416 \text{ or } \frac{22}{7}$$

and the diameter of a boiler should be measured (for development) to the centre of thickness of the plate.

The intersection of cylinder with cylinder is given at Fig. 315, and the method of developing the plane surface: A and B representing a dome and boiler respectively. Taking the dome in plan, divide the circle into, say, twelve parts, and number as shewn. Calculate half-dome circumference, and lay out as at C D, dividing into six parts by vertical lines. Project lines up

from plan to meet boiler circumference, and carry these along horizontally to cross the vertical lines at C D; the serpentine curve, being then traced through the numbers obtained, will represent the developed intersecting line. This may be repeated on the second half of the plate, and allowances made for flanging and welding. The boiler hole is developed by stepping-off the three distances, h, h_1, and H, with dividers, and measuring them from the vertical centre line in plan to give a, b, and c respectively, the remaining four segments being symmetrical. The length of plate is found by calculation.

Intersections of oblique cylinder with plane, or cone with cylinder, are rarely required; but cone with plane is sometimes necessary, as in funnels for American locomotives, or conical flues such as that shewn at L, Fig. 310. The latter has been chosen as an example, and the form of plate developed at K, Fig. 315, J being the finished flue. The drawing J having been made, the outer lines are produced to meet at f, and the dotted circles struck, with $g\,f$ and $j\,f$ as radii. Upon these are measured the circumferences at d and e respectively, and allowance made for welding and flanging.

If the set-squares at hand be not long enough, the marker-off should be able to set out a right angle by the measurements of three sides of a triangle, it being easily remembered that the proportions 3, 4, 5, for base, perpendicular, and hypotenuse in turn, will serve his purpose, as can be proved by the 47th proposition of Euclid's first book, thus:

$$3^2 + 4^2 = 5^2 \qquad \text{or} \qquad 9 + 16 = 25$$

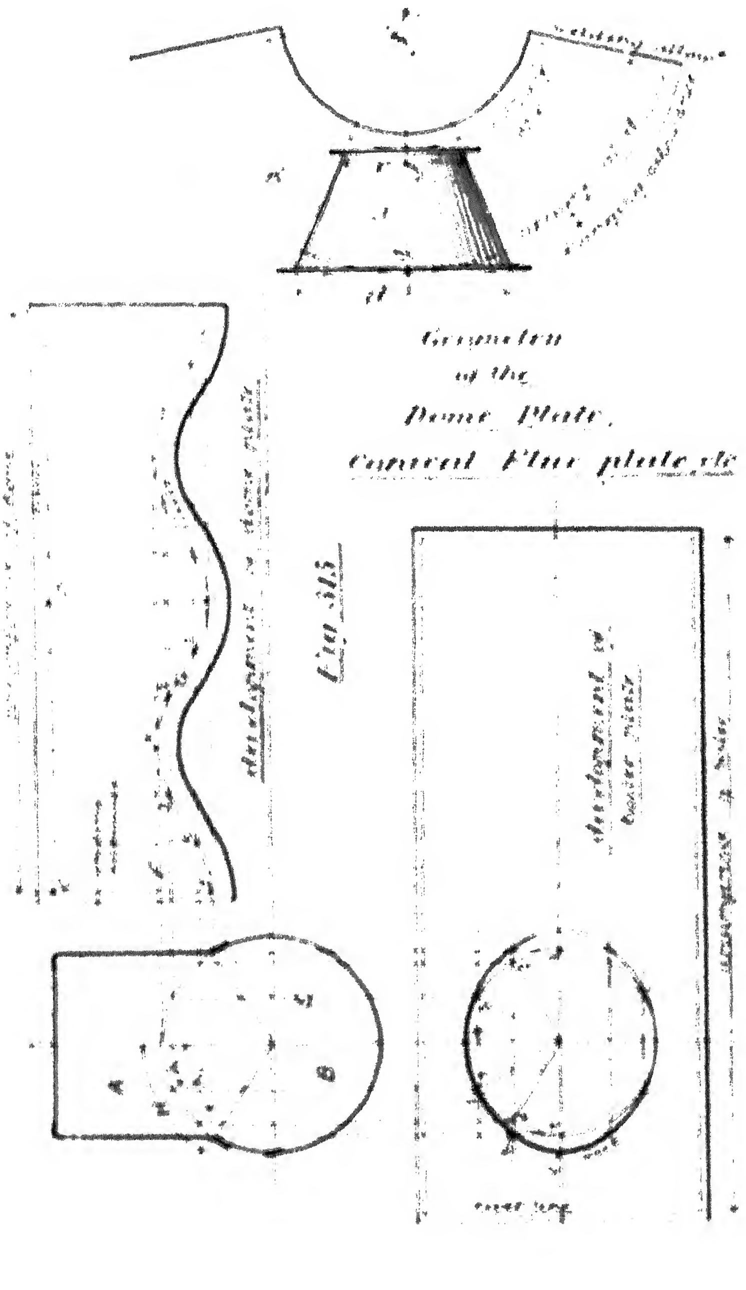
The length of arc, chord being known, is sometimes required, and may be obtained as follows:—

Let c = the half chord.
r = radius of arc.
a = half the angle subtended by the arc.

$$\text{Then } \frac{c}{r} = \sin a.$$

The angle a being found from a table of sines,

$$\underline{\text{Length of Arc}} = 2 \times \frac{a}{360} \times 2\pi r = \underline{\cdot 0349 a r}.$$

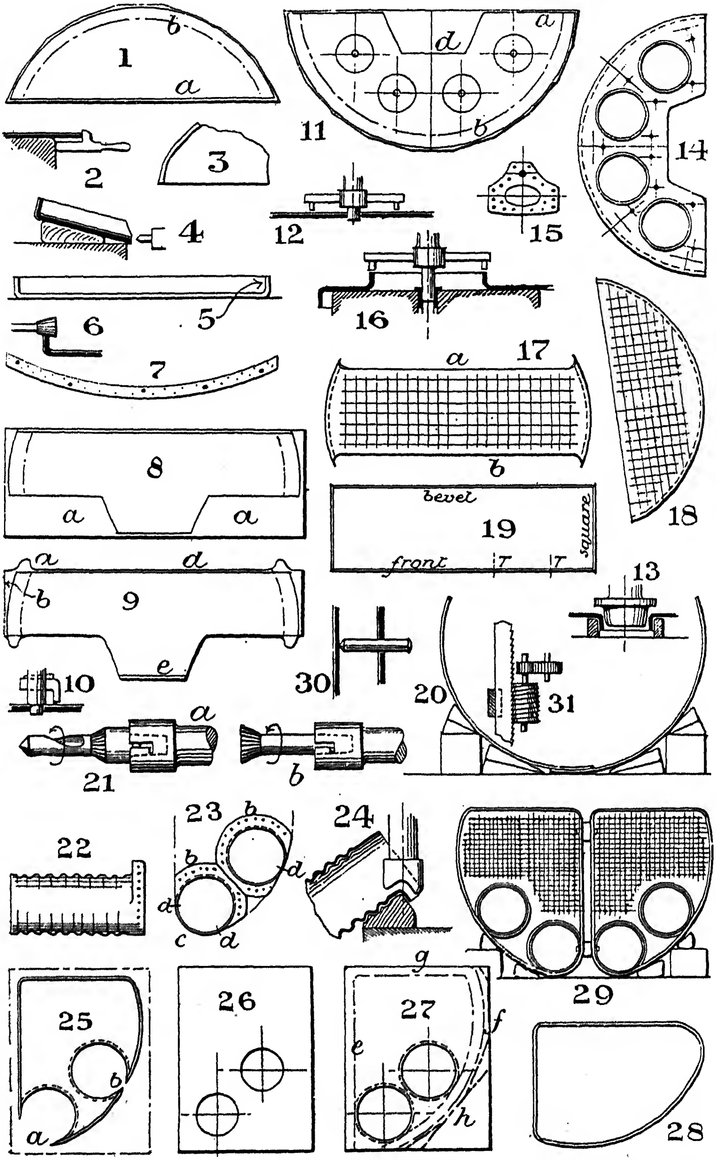

Geometry
of the
Dome Plate,
Conical Flue plate etc
Fig. 313
development of dome plate
development of base plate

Setting-out a Marine Boiler.—We are now in a position to detail the method of setting out the plates and putting together of any form of boiler previously described. Taking first the *Marine Boiler:* the Draughtsman must make a list of the plates required, taken from the drawings, for ordering purposes, giving each a marginal allowance, which will vary from $\frac{1}{4}$ in. to $\frac{1}{2}$ in. all round according to thickness of plate. This is necessary, for the shears at the Rolling Mill leave a rough edge and distress the plate. Referring to Fig. 311, the plates N, P, Q, and S would be ordered as 'sketch plates'; coming in roughly sheared to the shape: G and X might also be cut down at the mill; but the remaining plates would be ordered to the nearest rectangle. Care must be exercised to remember flange or lap allowances. The Fox tube is rolled by special machinery, so must also be 'ordered out.' When received, it must be carefully gauged at every ring, and if found to be more than a $\frac{1}{4}$ in. oval, must be rejected.

Supposing all the plates have been received, we will refer to the sketches in Figs. 316 and 317, taking the Front plates first.

I. *Front Stay Plate.*—This is received roughly sheared, as at 1. It is painted with whitening and marked off to drawing, as shewn by dotted lines, keeping *a* near the edge to avoid much planing. Then the curve *b* is cut out by band saw to give an edge for the flanging gauge 2. Flange to gauge, by the progressive method, Fig. 290, the ends being set as at 3, by the horizontal ram. Being now considerably strained, the plate is placed in a furnace, and uniformly heated to a dull red heat; on removal it is laid on a flat table, and straightened by wooden hammers, then allowed to cool slowly. The edge *1a* is next planed on the machine in Fig. 285, and a bevel given by setting as at 4, the angle being 1 in 8; often this is given to outer edges only. The long edge is planed with a stroke the full length, and the flange 5 with short strokes, the position of the stops Q, Fig. 285, being altered for the purpose. The flanged edge is milled as at 6, with a conical cutter, to obtain caulking inclination, a suitable table being provided to give a curvilinear feed.

The rivet holes are now drilled to the extent of one in every six, *measured along the pitch line*, for use in holding the plates together while drilling in position. In this case the holes along



Setting-out a Marine Boiler. Fig. 316.

1*a* are to be marked from the tube plate; but those along the flange are obtained by laying upon the latter a *very thin* steel strip 7 prepared with marked holes, and of the exact length of the flange. After marking through, the flange holes may be drilled in a Horizontal Drill, and the plate holes in a Radial Drill. The holes for the stay bolts are marked off to dimension, as shewn at U, Fig. 311, and drilled with clearance for the bolt.

II. *Front Tube Plate.*—The Plate is first marked as at 8, and part 8*a* cut out by Band Saw. The pieces 9*a* are next drawn out to a tapering wedge as at S_1, Fig. 311, after which the parts 9*b* may be removed by Band Saw. Flange 9*b* to gauge; anneal and straighten. Plane edges 9*d* and 9*e* to a bevel, trimming the corners by chipping, and mill the flanged edge as before. Set out all the plate rivets as at W, V, and Y, Fig. 311, and the tube holes. Prepare a steel strip the exact length of the flange, and pitch the rivets upon it; then mark through to the flange one rivet in every six, leaving the corner rivets. (N.B.—It should be remembered that the corner rivets, where three plates overlap, are always better drilled absolutely 'in position.') Now drill all the plate rivets under a Radial Drill, and the tube holes at the same time, making first a small guide hole for the pin drill 10. The stay tube holes are made to tapping size, and the other tube holes to gauge. The flange holes are drilled in a Horizontal Drill, and the stiffening plates (V, Fig. 311) marked from the tube plate and drilled separately.

III. *Bottom Front Plate.*—Whiten the plate as before; draw centre line, line 11*a*, and strike curve 11*b*. Set out the centres of the furnace holes, and strike a circle on each, smaller than the flue by the flanging allowance. Drill a small hole for drill steadiment at the furnace centres, then lay the plate on the drill in Fig. 291, and take out the large hole by the trepanning tool 12. Heat and flange, as at 13, each of the furnace holes, and after cooling lay on the marking table to test the original lines, which have drawn a little; so the curve 11*b* must be re-struck, and cut by band saw. Flange to gauge, including the setting of the flange ends; anneal and straighten. Mark out line 11*d* and cut out with band saw; plane also the edge 11*a*. If possible, give the bevel at *d* when cutting, but if that is not convenient, finish

by milling or chipping. Mark the flange rivets, one in six, with a special steel strip, and the rivets along the seam *a d*, one in six, from the tube plate. Set out the centres for stays Y Y_1, Fig. 311, and mudholes Z Z, as shewn at 14. Next prepare the stiffening plates 15 by marking out, sawing, cutting the oval hole by the special method shewn in Fig. 291, and drilling the rivet holes. Place the stiffening plates in position, and mark through all their holes; then drill all holes by a Radial Machine, and cut the mudholes by the appliance in Fig. 291. The edges of the furnace flanges are tooled in the same machine by fixing the plate horizontally on the table and revolving the tool Q Q, as at 16.

IV. *Top Back Plate* is prepared in the same manner as I.

V. *Back Middle Plate.*—This must be lined out as at 17, with *a* and *b* parallel, and the curves struck. The rest may be understood from II. After planing *a* and *b*, and setting out the stay holes, the latter are left to be drilled till all are bolted together.

VI. *Bottom Back Plate* (18) is treated in the same manner as I., but the stay holes are all drilled in position, as in last example.

VII. *Front Ring Plates.*—There are three of these, all equal in length. They are lined, as at 19, with long set squares, then planed, the long edges to a bevel, and the short edges square; next taken to the Rolls, Figs. 286–7, and put through in the manner previously described. But many Marine firms prefer to work with Vertical Rolls, believing that besides supporting the weight, the curve is obtained more squarely with the long edge. In finishing the short edge, a greater pressure is given to secure accuracy of curvature, and partially avoid the necessity of bending with hammer. Now mark off the rivet holes to suit those already drilled in the flanges of the Front and Back plates. To this end the steel strips are again used, and, being very thin, do not differ appreciably in their outside and inside circumferences. The positions of joints T T must be found with relation to the butt joints F F (Fig. 311), and the centres of T T marked upon the front long edge of the ring plates. Then the steel strips are applied, and the holes marked to correspond with the flanges. Of course these strips must be all carefully numbered, to avoid

mistaking the one for the other. The rivet holes, one in six, for the back long edge must be set out so as to bring the joints F and B (Fig. 311) into exact relation with each other. B B are therefore marked upon the Front Plate, and two methods occur by which the intermediate holes may be traced: one involving the use of the thin strips, and the other being the placing of one plate upon the other, on blocks as at 20. The latter method seems preferable, because *all* the holes may be marked on the back edge of Front Plate, one in six drilled, and then traced through to the Back Plate, VIII. The manhole is next marked off, with its rivet holes, but is not cut out till in position. The butt strap is prepared by planing; heating and pressing to correct curves between dies; then marking off all holes, but drilling only three on each edge. It is next applied to the plate, these holes marked through and drilled.

VIII. *Back Ring Plates.*—These are also in three, and of equal length. They are marked as in the last example, and if care be taken, the horizontal joints of the plates II. and V. will be in line with each other. This is a necessity, so it is advisable to keep the vertical centre line of the boiler well in view, on all these plates I. to VIII., during the whole of the marking off.

We may now bolt together the whole of the shell plates through such rivet holes as have been drilled, and place the boiler upon the cradle A A, Figs. 293-4, Plate XIII. The drill spindle is adjusted as there described, and all the holes in the ring plates drilled right through. There are two principal forms of rivet holes required, as shewn at 37 and 38, the former being for machine and the latter for hand-riveting. In 37 the arridge is just taken off, while 38 requires a deep countersink, but both may be given by the tools 21 (*a* and *b*). 21*a* is applied from the outside, and withdrawn when the hole is finished. 21*b* is then passed through from the inside of the boiler, and fastened in a special slot as shewn. Its teeth cut left-handed, so the machine need not be reversed, but the backward feed is given by hand, and the depth gauged by a mark on the drill. All the shell rivets are like 37, excepting those in the back flange, and even they may be machine-riveted, as will be shewn. The manhole is taken off by drilling holes round its circumference close together, then

finished by chipping. The bolts being clamped, their holes are also countersunk, being first rimered to ensure exact correspondence. The rivet holes both at front and back of boiler are next drilled by placing the latter on a cradle, which allows the flat plates to stand vertical, and face four drill standards supporting horizontal drills on suitable saddles. The boiler joints being truly level, the rivet holes may be easily drilled, as well as the stay holes in the back, the latter being made to tapping size.

IX. *The Furnace Tubes* (22 and 23) are usually obtained rolled, flanged, and cut to correct shape, an allowance being left at front end for turning. They may be flanged, however, under the machine in Fig. 290, as shewn at 24, using special dies. Mark off all the flange holes, as at 23, and drill *all* those at *b*, one in every six at *c*, but none at the corners *d*.

X. *Combustion Chamber Throat Plate.*—This is flanged to the shape shewn at 25. A rectangular plate being procured, the centres of the furnaces are found as at 26, a hole trepanned, and the flanging of the throat done at one heat, as at R, Fig. 290. The rest of the plate is lined as at 27 and the corners cut, the sides *e*, *f*, *g*, and *h* being flanged progressively until the whole fits a cast-iron block or template. This is of course an operation involving great care. Now the portions 25*a* and 25*b* are sawn out, finishing the plate with the exception of the taper ends, which are drawn out by heating and hammering on the cast-iron block. After milling the flange edges, the rivet holes 23*b*, connecting with the Fox tube, are marked from the latter, and drilled separately; and the flange holes carefully spaced out by reference to the top corners and the furnace centres, but only one in every six drilled now, and none through the taper portion.

XI. *Combustion Chamber Back Plate* (28).—This must be lined out and flanged progressively to fit a cast-iron block, and the flange edge then milled. The stay holes are drilled in position.

XII. *The Cover Plates* for the Combustion Chamber are now edge-planed, rolled, and bent hot with hammer, until they exactly fit the flanged plates, as shewn in Fig. 311. There are three of these plates, one for the roof, and one for each side; and the holes already drilled in the flanged plates must be traced through

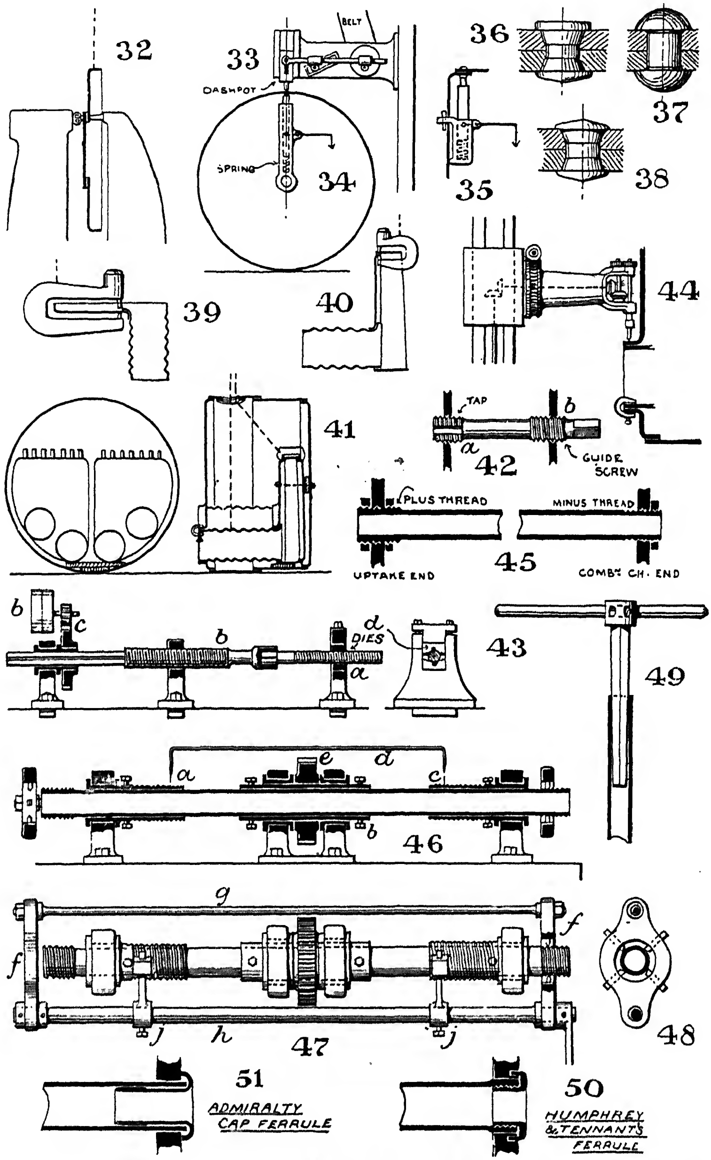
upon them. The inner laps at the joints *b b b* must of course be tapered, but no holes are yet drilled there, or through any of the tapered pieces.

Fix all plates of both chambers, including Fox tubes, with temporary bolts; and, laying each upon its back, drill with Horizontal Drill *all* the rivet holes, as spaced on the cover plates. Mark out and drill to tapping size the stay holes in the mid cover plate of *one* chamber only, and drill also the holes for the girder-stay bolts. Set up both chambers in position as at 29 by bolting through the rivet holes, and blocking below. Obtain level position with great exactness, then draw horizontal tube centres by squaring from the roof, and the vertical lines from the middle plates. They are afterwards drilled to correspond with II. The mid stay holes are marked from one chamber to the other by a punch 30, of the same diameter as the tapping size of the holes, and afterwards drilled by Horizontal Drill.

The Girder or Roof Stays are now cut out by band saw, being clamped together, and are next fitted to the roof, as shewn in Fig. 311.

The Band Saw is a very useful tool, but requires some attention to keep it keen. The tool at 31 is a roughened steel helix, rotated by gearing to sharpen the saw teeth as the band is advanced.

Riveting the Boiler.—The Front and Back Plates may now be put together in a Fixed Riveter as at 32, and the ring plates attached by the same machine up to the condition L, Plate XVI. But the Back Plate must either be put in by hand or semi-hand process, or by the machine at P, Plate XVI. The combustion chamber (after riveting up) is first inserted, and laid loosely within the shell. Then, if hand-riveting be used, the rivet will appear like that at 38, the flat finish being obtained by very quick consecutive blows from riveting hammers used by two workmen, while a third 'holds up' a cupping tool within the boiler. The hammering is continued on both sides after the rivet is cold, as a sort of caulking. A pneumatic hammer is employed in some works, as at 33, where a lever vibrates from a crank plate driven by a belt, while the hammer end is provided with a pneumatic dashpot or cushion, giving a finish like 36. The



Setting-out a Marine Boiler Fig. 317

holding up may be obtained as at 34 or 35, by pressing on the levers there shewn.

But the boiler may be finally closed by machine, using the methods at P or M, Plate XVI. The former is adapted to internally flanged boilers, the tube plate being cut in three pieces at the stiffening plates. After the flange has been riveted, the various tube plate rivets may be closed by the usual Lever Riveter with long arms, dropped in through the furnace holes. The best result is obtained by a boiler designed as at M, Plate XVI., and this should be employed whenever the ship designers permit it.

The *Combustion Chambers* are put together as at 39 and 40, but the back plates are riveted by hand, with rivets like 38, unless the flanges be made as at M, Plate XVI. The chambers and furnaces are next put within the boiler shell, and the latter closed. They are slung as at 41, carefully blocked and bolted in position, then clamped at the front. Placing the boiler on a cradle, before horizontal drills, and on the machine in Plate XIII., drill the stay holes through into the Combustion Chamber to ensure exact alignment for the screw threads. All stay holes, including those between the chambers, are now tapped, as at 42, by a tap whose threads *a* and *b* are continuous.

The *Screwed Stays* are prepared on the machine at 43. Stay *a* is coupled to spindle *b*, which revolves by gearing *c*; screw *b* has the same pitch as die nut *d*, and prevents the formation of unequally pitched threads on the stay by 'drawing' or uneven pressure. The stays, having a square on their ends, are now placed in the boiler with a wrench, a nick being first turned at each end to represent their exact lengths; so that having been advanced to correct position, a sharp twist will break off the surplus material. Nuts are now added, and the stay ends trimmed up.

The boiler being still upon its cradle, the rivet holes at the furnace mouth are set out and drilled by the machine at 44. The drill bracket may be revolved on a horizontal axis by worm gearing, and this, coupled with the rotation of the boiler, will enable us to drill all round. The riveting-up is shewn at K, Plate XVI.

The Stay Tubes must next be screwed. They are either formed with a *plus* thread at one end and a *minus* thread at the other, as at 45, or both ends may have a plus thread. The first involves less labour, while the second is stronger. The tapping machine is shown at 46, 47, 48. First the tube is cut to length, and placed within the bushes *a b*. After *a* and *b* have been adjusted till the trammel *d* (whose length represents an exact number of threads) fits their thread grooves, the set screws are tightened, and the spur wheel *c*, being rotated, will also turn the tube and the bushes. At 48, the end view of *b*, are seen two screwing and two chasing tools, the one pair being withdrawn while the others are in operation, and the two pieces *f f* are united by a back rod *g* and a shaft *h*. *h* is again provided with two arms *i i*, which hold copper dies resting on the bushes *a* and *c*. It follows, therefore, that when the machine is in operation, the tube turns, and the screwing tools advance to cut the screw on the tube ends of the same pitch and with a perfectly continuous thread, as obtained by means of the adjusting trammel *d*.

The Tube plates are next tapped. A short tap of the usual form is used for the front plate, but after that is done, a long tap like 42 is inserted to screw the back plate, *a* being the tap and *b* the guide screw. Of course, as before, the two threads must be continuous. The stay tubes are inserted with a square drift and wrench 49, while the plain tubes are expanded at the uptake end and ferruled at the opposite end, then cut off by the tool at Fig. 306. The ferrules at 50 and 51 are found most effective for marine work.

The Manhole seating is now flanged. A ring is cut out of a solid plate by trepanning, and then bent over blocks by hammers to the shape *m*, Fig. 311. Of course this occupies both time and labour, and probably a method of machine flanging might be suggested. The stiffening plate being also provided, both pieces have their holes marked from the boiler, are then drilled, and riveted to the shell. The longitudinal stays are prepared and screwed, their washers turned, and all bolted up in place.

The seams and rivet heads are finally caulked, and the boiler tested:—(1) by hydraulic pressure, to about twice, and (2) by steam pressure, to about 1½ times the working pressure.

Setting out other Boilers. [illegible] details for a Marine Boiler [illegible] some little variation being necessary [illegible]

The *Locomotive Boiler* [illegible] plate [illegible], the throat plate [illegible], the firebox [illegible] plate [illegible], and the [illegible] heat, between [illegible] then the edges are [illegible] holes and [illegible] are [illegible] and [illegible]. The [illegible] plates are [illegible] the dome [illegible] that the [illegible] plates heated and drawn [illegible] taper where necessary. These plates are next rolled to [illegible], as shown in plan at [illegible]. The firebox shell [illegible] must have its developed outline marked out upon the unrolled plate, including the [illegible] at m, and the hole at [illegible] cut out by hand saw or by a [illegible] of punching and shearing, finishing under a vertical [illegible] in either case. The remaining edges are planed, and [illegible] with a bevel for caulking. The hole at [illegible] is punched in the same manner as [illegible], with the piece left in as a support during rolling. The crown plate may now be bent carefully to fit the flanged plates. Heat and taper all [illegible] of plate ends that have to lie between two other thicknesses. Prepare the firehole ring [illegible] and foundation ring [illegible] by forging from wrought iron bar and welding; forge also the girder stays [illegible] and the safety valve seating [illegible], both being steam-hammer work. The last is bent to the shell curve, then planed underneath and surfaced on top. Weld the mudhole bosses [illegible] on the plates, either by roughing the two surfaces to be joined, or by shouldering down the boss and riveting through on the inside. Bend the angles [illegible] and ring [illegible] in the machine at Fig. [illegible], the ring being welded with a glut piece. Forge the sling stays and pins at [illegible], and cut off the angles [illegible] and [illegible] with circular saw. The dome plate is rolled into a cylinder and welded, then the flanges at top and bottom are formed by heating and bending over special blocks with wooden hammers, and the stiffening piece is cut from solid plate.

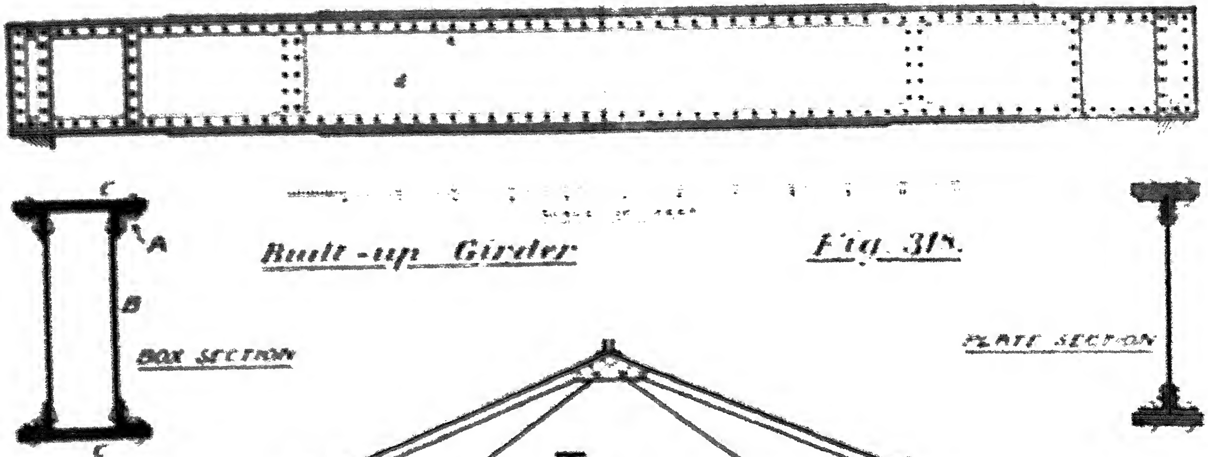
All the plates are now prepared, and must next be marked off for drilling. First the tube holes are carefully lined on the two tube plates, and cut out by pin-drills in a radial machine. Then the *outer* plates may have their seam rivets spaced out, and one in every six drilled, always omitting the corner holes, or those where three plates overlap. The various parts may now be bolted together, and all the rivet holes drilled and countersunk. Thus K and Q being connected, the tube-plate rivet holes may be done in a radial drill; adding plate J, the circular seams may be drilled, as described at page 308, including also the holes in the dome-hole stiffening piece, and those for the smokebox plate. The dome flange is marked from the boiler and drilled separately. Bolting H to J, the firebox shell may be drilled round its circumference in like manner, but those on the flat sides would be done under a radial or multiple drill, the latter being preferable. The barrel is now disconnected from the firebox shell, and the firebox bolted to the latter; then the whole shell placed on the lower table of the Multiple Drill in Plate XIV., and the stay holes drilled right through both plates to secure accurate alignment. All remaining holes are now made, such as those for the angles T, W, and P; for the seatings V and M; for the palm stays at S_1; and for the guide stays at E.

The operation of riveting is clearly shewn at Plate XV. The barrel and shell are closed by fixed riveter at A and O, and the firebox partly by O and partly by portable riveter. Then the smokebox plate and the firebox are each fastened to the boiler shell by portable machines, as shewn at G, L, and H. Finally, the dome may be riveted as at P, so there is no occasion for hand work on any part of the boiler. Note that the angles W, T, and P must be riveted before the firebox is put in.

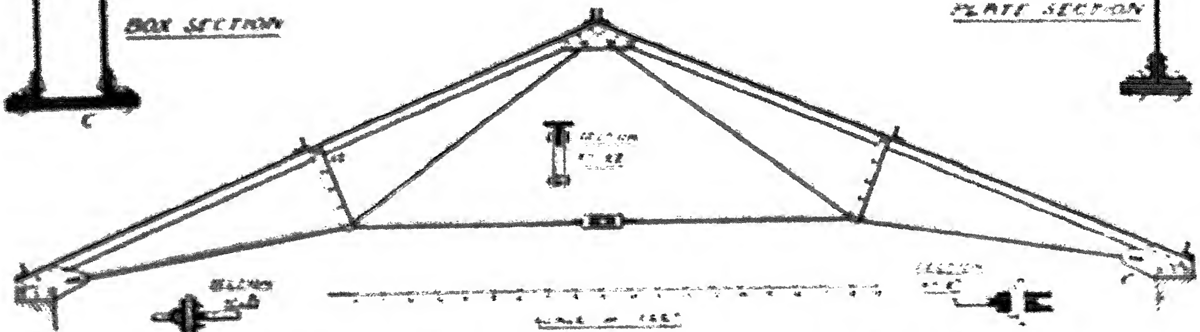
The tubes are fixed by expanding at the smokebox, and beading and ferruling at the firebox end, using the tools in Figs. 304 and 305; and the smokebox ends of the tubes are then cut off by the tool in Fig. 306. The screwing of the stays will be understood from the marine example, but in this case their ends are riveted over by hand after fixing. The mudholes are tapped to suit the plugs, the guide stays screwed into place, and the steam pipe M expanded into the plate. The boiler is lastly caulked throughout and tested.

The *Lancashire Boiler* (Fig. 310) may be next considered shortly. The back and front plates are turned, trepanned, and drilled throughout, with the exception of the rings *a*, *b*, *p*, and *q*, these being marked afterwards from the angles. The shell plates are prepared as before and drilled in position with axis vertical, two by two. The angle ring *a* is also drilled for the shell, and the holes at *b* for the flange; then all are riveted together in batches of three, with a fixed machine, and the batches connected by hand, or by the method at 34, Fig. 317. Next the flue plates are rolled, welded, and flanged as at 24, Fig. 316; turned on machine, Fig. 291; drilled in position by machine, Fig. 292; and riveted together, with caulking strip between, by a portable riveter. The plates *j* and k_1 are to have the angle rings *p* and *q* attached, but the plates themselves are first *bolted* to the other tubes, and the whole tested with a long wooden lath to see if it will make up to the same length as the boiler shell; then the end tubes turned down accordingly. The general straightness of the tube should be tried during riveting, and adjusted by varying the thickness of the caulking strip. Now the rings of holes—*a*, *b*, *p*, *q*—may be marked on the end plates. First the holes at *p*, *q*, and *a* are marked and drilled. Then the shell is laid horizontally, the flues blocked up in place, the back and front plates put on, and bolts put in the rings *a*, *p*, and *q*; when the holes in the shell at *b* may be traced through to the flange. Removing the back plate to drill the flange holes, the gusset stays are prepared with their angles riveted on, and are placed within the boiler. The back plate is once more bolted on, and the whole boiler lifted on to a trolley, which can be run under a radial drill, the latter being preferably hinged on a wall or shop pillar so as to be at a sufficient height while presenting no obstruction beneath. The holes *g*, *h*, and *f* are cut out by drilling, and those in the shell, for the gusset stays, lined out by squaring from the end plates, then drilled. Entering the boiler, the workman places the stays in position, and marks off the remaining rivet-holes in the end plates. Removing the back plate again, the gussets are taken away to drill, then all are replaced for riveting.

The gussets, the flange *b*, and the rings *p* and *q*, must be riveted by hand, but the ring *a* may be done by machine.



Built-up Girder — **Fig. 318.**



Iron Roof-Principal — **Fig. 319.**

Prepare the longitudinal stays and manhole seating; put in place, with fittings; and test the boiler as before.

The *Vertical* and *Water-tube* Boilers present no further difficulty. Taking the first, the shell is built-up separate from the firebox and chamber. Machine riveting can be used for most of this work. But when putting together, the foundation ring is the only other part that can be done by machine; all the rest is hand work. The tubes are expanded into the tube plates as before.

The *Water-tube Boiler* (*p.* 338) has its tubes cut to length, and expanded into the headers; the chambers *a b* flanged and welded; while the making of G will be understood from previous descriptions.

As further examples of Plate Work, we illustrate a *Girder* at Fig. 318 and a *Roof Principal* at Fig. 319; but these are simple in comparison with boilers, as far as their practical construction is concerned. The Box Girder has its plates and angles sheared to dimension, the holes then marked off, punched, and rimered in position. The angles A and web plates B are first riveted, and next connected to the booms C C; so it will be clear that no hand-riveting whatever is necessary. The Roof Principal needs no explanation. The first application of portable riveting to bridge erection was made by Mr. Tweddell in 1873, on the Primrose Street Bridge, London.

Ships are now built of steel plates and angles, whose dimensions are carefully got out by the draughtsman in the first place. Much more drilling is now done than formerly, though a considerable amount of punching prevails, and the plates are usually sheared. The keel and framing are first erected, and the plates then adjusted and marked from these. As regards the riveting up, nothing could shew this better than the diagram at Fig. 301. Of course there are many plates too long to be reached by the machine, but this diagram shews what an extraordinary amount of work can be performed by these wonderful 'Portable Riveters.'

PART II.

SYNOPSIS OF LETTERING ADOPTED IN THIS PART.

CAPITALS.

A	Area in square feet.
B_m	Bending moment. [efficient of discharge.
C	Modulus of transverse elasticity in lbs. per sq. inch: Co-
D	'Larger diameter' in inches.
E	Modulus of direct elasticity in lbs. per sq. inch.
F	Total stress in tons: F° = Fahrenheit.
F^{lbs}	Total stress in lbs.
F_n	Tractive force to overcome friction: in lbs.
G	Weight of a cubic foot of water: Centre of gravity.
H	Height or head in feet: Total heat.
H.P.	Horse power per min. = 33000 foot pds.
I	Moment of inertia $\{\Sigma\,(\text{area} \times r^2)\}$.
J	Joule's equivalent.
K	Modulus of volumetric elasticity in lbs. per sq. inch.
K_p	Specific heat of a gas at constant pressure; in foot lbs.
L	Length in feet. [K_v = ditto at constant volume.
L_h	Latent heat. [C_p and C_v = same quantities in heat
M	Poisson's ratio. [units.
N	Number of revolutions per min.
O	Coefficient of bending stress. [in lbs. per sq. foot.
P	Total pressure in lbs.: Effort, or force applied: Pressure
P_{tons}	Total pressure in tons.
Q	Concrete of formula for struts $= \dfrac{\pi^2 E I}{l^2}$: Water discharge
R	Radius in feet. [in cub. ft. per sec.
R″	Larger radius in inches.
R_t	Reaction at supports. [heat.
S	Range of stress variation in Wöhler formula: Sensible
T	Number of teeth.
T_m	Twisting moment.
T_n	Greater tension in belt or rope.
T°	Final temperature in heat mixtures.
U	Work put in.
V	Velocity in feet per min.: Volume in cub. ft.
W	Weight or load in tons: Resistance, or force removed.
X	Number of bolts in flange coupling, cylinder cover, &c.
Y	Concrete of formula for beam deflection $= \dfrac{w\,l^3}{48\,E\,I}$
Z	Modulus of section (in bending).
Z_t	Ditto (in twisting).

SMALL LETTERS.

a Area in sq. ins.

b Breadth in ins. [constant.

c Contraction coefficient for gun coils; cylinder clearance;

c_v Coefficient of velocity.

d Diameter in ins., or 'smaller diameter.'

e Base of Napierian or hyperbolic logarithms = 2·718.

f Stress per sq. in. in tons (generally); acceleration in ft. per sec.

f_t, f_c, f_s, f_b Stresses in tension, compression, shearing, and bearing, respectively, in tons sq. in.

f^{lbs} Stress per sq. in. in lbs.

f_l Lateral stress.

f_r Modulus of rupture (in bending).

f_h Hoop stress.

g Acceleration of gravity in ft. per sec. = 32·2 at London.

h Height in inches.

i Intermediate radius of thick cylinder.

j

k Pitch of bolts in terms of bolt diameter.

l Length in inches.

m Mass in lbs. $= \dfrac{w}{g}$

n Number of revolutions per sec.

p Pressure in lbs. per sq. in.

p^{tons} Pressure in tons per sq. in.

p Pitch of screw or of riveted joint.

q

r Radius in ins., or 'smaller radius.' Ratio of expansion.

s Side of square in ins.; specific heat.

t Thickness of plate; time in secs.

$t°$ Temperature, or rise of temperature, in deg. F.

t_1 Lesser tension in belt or rope.

u Work given out.

v Velocity in feet per sec.; volume in cub. ins.

w Weight or load in lbs.

w' Width of one link in rivet calculations.

x Coefficient in Wohler formula. [twisting).

y Distance of furthest fibre from neutral axis (in bending or

GREEK LETTERS.

Small Letters.

α (*alpha*).	Coefficient of linear expansion in degrees Fahrenheit: various angles.
β (*beta*).	Various angles.
γ (*gamma*).	Ratio of $\frac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}}$.
δ (*delta*).	Deflection per inch length: δ^{feet} = ditto per foot.
η (*eta*).	Efficiency.
θ (*theta*).	Angle of torsion.
κ (*kappa*).	Coefficient of jet contraction.
μ (*mu*).	Coefficient of friction or tangent of friction angle.
π (*pi*).	$3\cdot1416$ or $\frac{22}{7}$: ratio of circumference to diameter.
ρ (*rho*).	Radius of curvature in bending: coefficient of resistance.
σ (*sigma*).	Various angles.
τ (*tau*).	Absolute temperature in F°.
ϕ (*phi*).	Angle of friction: entropy.
ω (*omega*).	Angular velocity.

Capitals.

Δ	Total deflection in inches.
Δ^{ft}	Total deflection in feet.
Σ	'Sum of.'

SIGNS.

$\propto$	'Varies as.'	$\parallel$	Parallel to; with fibre.
$>$	Greater than.	$+$	Across fibre.
$<$	Less than.		

PART II.—THEORY AND EXAMPLES.

CHAPTER VIII.

THE STRENGTH OF MATERIALS, STRUCTURES, AND MACHINE PARTS.

OUR intention in this chapter is to treat of the cohesive strength of the materials used in Mechanical Engineering, of practical testing to obtain strength constants, and of the use of the latter in proportioning machine parts, so far as may be done.

Load is the total effect on the structure of the external forces, and may be 'dead' or 'live,' concentrated or distributed (*see pp.* 391 *and* 438).

Stress is the cohesive force within the material called into play to resist the load. (*See Appendix III., p.* 920.)

Strain is the deformation produced by the stress.

Kinds of Stresses.—Only three *simple* stress-strain actions are possible: *tension* (pulling), *compression* (thrusting), and *shear* (cross-cutting). Bending is a mixed action, and local compression produces a *bearing* stress. Fig. 320 shews the distortions and fractures produced by these various stresses.

Elasticity is the property of regaining original shape and dimensions after distortion; very apparent in an *elastic* body, but scarcely perceptible in a *rigid* one. In 1676, Hooke propounded the law 'ut tensio sic vis' (as the tension, so the strain), meaning that stress and strain are proportional, if within the elastic limit of the material.

Limit of Elasticity.—A bar being subjected to an increasing stress (of any kind), will receive also a proportionately increasing strain (of the same kind) until the *elastic limit* is reached, after which the strains increase more rapidly than the stresses till rupture occurs. Shewing this by a diagram, Fig. 321, o is an

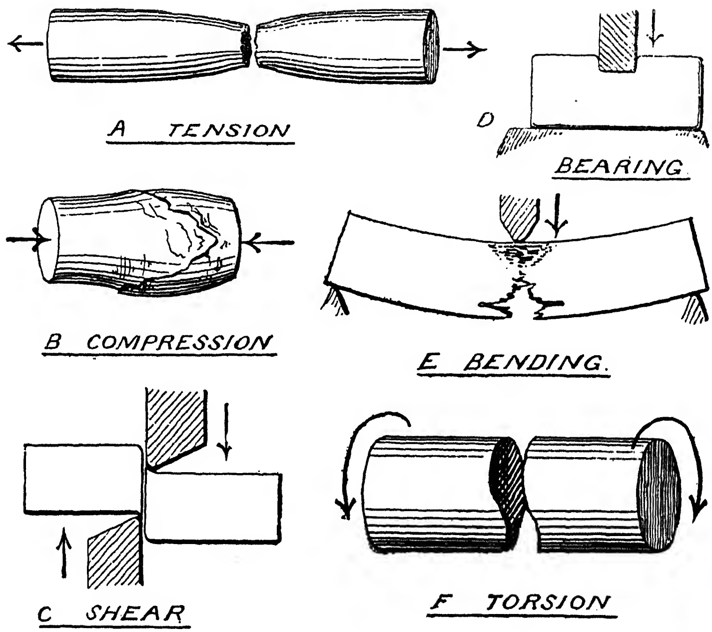


Fig. 320 Distortions & Fractures

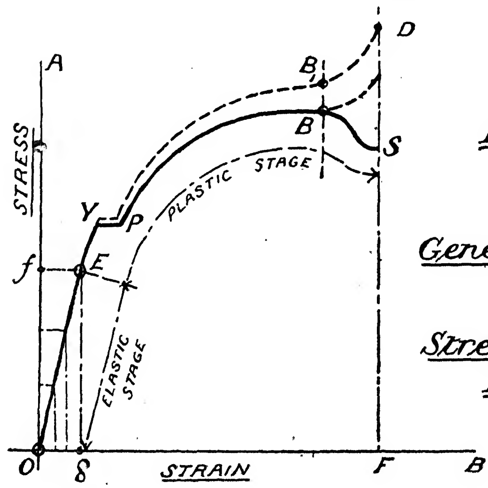


Fig. 321

General form of Stress-Strain Diagram.

origin from which stresses are measured along O A, and strains along O B. E is the elastic limit and O E is a straight line, shewing proportionality of the co-ordinates. Plasticity begins at E, and (assuming a case of tensile stress) increases in perfection up to B, the curve being interrupted at Y, the *yielding or breaking-down point* (or commercial elastic limit), while the lowering at B S indicates rapid contraction of sectional area at rupture (*see* A, *Fig.* 320). If the stress be compressive the material enlarges in diameter after B is reached, and thus becoming stronger, the curve rises thence instead of falling: neither is the yield point observed.

If W = load in tons at B, a = original area, and a_1 = contracted area:

$W \div a$ = stress per sq. in. estimated on original area.

and $W \div a_1$ = stress per sq. in. estimated on contracted area.

The first is used commercially, and is shewn at B, while the latter, the strictly scientific result, is given at B_1; and the plastic curve is thus corrected. The curve from B to S is not considered reliable.

Compressive stresses do not materially distort the specimen up to B, so the curve requires no correction. The *primitive elastic limit* occurs at E, after which a *permanent set* is given to the bar. This limit may be altered artificially. (*See p.* 385.)

Modulus of Direct Elasticity, or Young's* modulus. (E) is a number giving the ratio of stress and strain within the elastic limit, and is practically the same for tension or compression

$$E = \frac{\text{stress sq. in. in lbs.}}{\text{strain per inch length}} = \frac{f_t^{\text{lbs}}}{\delta_t} = \frac{f_c^{\text{lbs}}}{\delta_c}.$$

Modulus of Transverse Elasticity, or Modulus of Rigidity (C), serves similarly for shear action thus:

$$C = \frac{\text{shear stress sq. in. in lbs.}}{\text{shear strain per inch length}} = \frac{f_s^{\text{lbs}}}{\delta_s}.$$

δ_s will be understood by reference to Fig. 322, being the strain between two shear planes an inch apart.

Modulus of Volumetric Elasticity (K) compares stress and diminution in volume, thus:

$$K = \frac{\text{stress sq. in. in lbs.}}{\text{decrease in vol. per cub. inch}} = \frac{f_v^{\text{lbs}}}{\delta_v}$$

* Dr. Thos. Young, Foreign Sec. Royal Society, 1826.

TABLE OF ELASTIC MODULI (units being inches and lbs.).

Material.	E.	C.	K.
Cast Steel ...	30,000,000	12,000,000	26,000,000
Forged Steel ...	30,000,000	13,000,000	
Steel Plates ...	31,000,000	13,000,000	...
Mang. Bronze...	...	...	...
W.I. Bars ...	29,000,000	10,500,000	20,000,000
W.I. Plates ...	26,000,000	14,000,000	
Copper	12,000,000	...	24,000,000
Gun Metal ...	13,500,000	...	...
Cast Iron ...	17,000,000	6,300,000	14,000,000
Brass	13,500,000	...	15,000,000
Muntz Metal ...	14,000,000	5,250,000	...
Water	...	...	300,000

Mechanical treatment may raise these ratios: for tempered steel $E = 36{,}000{,}000$ and $C = 14{,}000{,}000$, while for rolled or drawn copper $E = 15$ or 17 millions respectively. (*See App. V., p.* 996.)

Poisson's Ratio (M) is a constant to determine the lateral effect of direct stress. If a bar, as in Fig. 323, be extended or compressed, it undergoes lateral contraction at A and expansion at B. Then, within the elastic limit:

$$\text{Direct strain} = \text{lateral strain} \times M$$
$$\delta_t \text{ or } \delta_c = \delta_l \times M$$

TABLE OF POISSON'S RATIO.

Material.	M.
Steel	3·25
Wrought Iron	3·6
Cast Iron	3·7
Copper	2·6
Brass	3·0

Nature of Shear Stress.—If the bar in Fig. 324 be subjected to elastic shear stresses, *s s* an equal pair of shear stresses

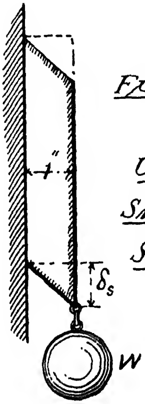
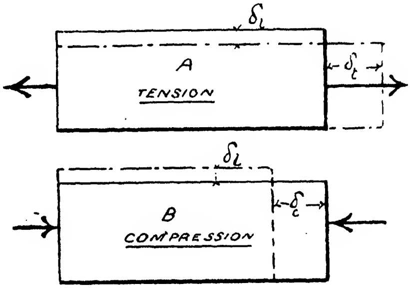
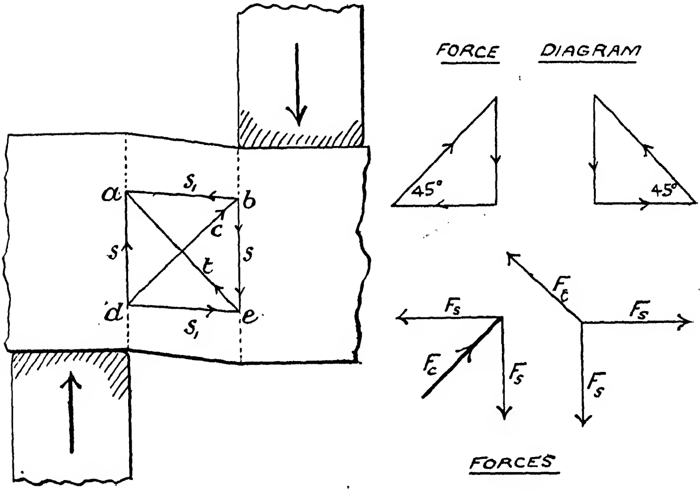


Fig. 322.

Unit Shear Strain.



Lateral contraction & dilatation by Poisson's Ratio. Fig. 323.



Nature of Shear Stress. Fig. 324.

s_1 s_1 will be induced on the cube *a b c d* (*p.* 859). Taking the corner *b*, the forces F_s F_s can be resisted by the force F_c which is wholly compressive, and the forces F_s F_s at corner *e* can be

similarly resisted by the purely tensile force F_t. A force diagram being drawn for each case,

$$F_c = \sqrt{2}\,F_s \quad \text{and} \quad F_t = \sqrt{2}\,F_s$$
$$\therefore f_c = f_t = f_s \qquad \textit{(See App. I., p. 755.}$$

Nature of Tensile and Compressive Stresses.—When a plain un-notched bar is broken by pulling, lines of cleavage appear on the surface, inclined at 45° to the axis; and the final fracture is cup-shaped. Compression fractures are also inclined at 45° and are often wedge-shaped. The evident deduction is that rupture takes place on shear planes in both cases, and that the three simple stresses are interdependent.

Work done by Uniform Forces.—The unit of work is a *foot-pound*, or *one pound exerted through a distance of one foot.* One pound acting through two feet, or two pounds through one foot, are each *two foot-pounds.* Hence:

$$\begin{aligned}\text{Work} &= \text{pressure} \times \text{distance}\\ &= \text{pounds} \times \text{feet} = \text{foot-pounds.}\end{aligned}$$

These forming a product may be represented by an area, for length × breadth = area, and A, Fig. 325, is therefore the diagram of work with uniform force:

$$\underline{\text{Work done}} = \underline{\text{pounds} \times \text{feet}} = \text{O X} \times \text{O Y} = \underline{\text{area A}}.$$

Work done by Variable Forces is shewn by diagram at B, Fig. 325. As the body moves from O_1 to 5, the pressure varies as $O_1\,X_1$, $2\,b$, &c. Now, work done between O_1 and 1 can neither be $O_1\,X_1 \times 1$ ft. nor $1\,a \times 1$ ft., but must be the average of these, or $O_1\,f \times 1$. In like manner the other dotted rectangles shew the work between the remaining intervals, and their addition,

$$\underline{\text{Area } O_1\,X_1\,b\,Y_1} = \underline{\text{work done.}}$$

Work done in Deforming a Bar is found at 1, Fig. 326. Divide O B into ten parts, and erect perpendiculars *between* the divisions. Measure the vertical ordinates in tons, then

$$\frac{\text{Total of ordinates}}{10} = \text{mean load in tons,}$$

and $\underline{\text{mean load} \times \text{extension}''} = \underline{\text{work in inch tons.}}$ (*See p.* 1065.)

Resilience is the work done in deforming a bar up to the elastic limit. 2, Fig. 326, is the diagram, where B A is the maximum elastic load, and O B the corresponding strain. (*See p.* 833.)

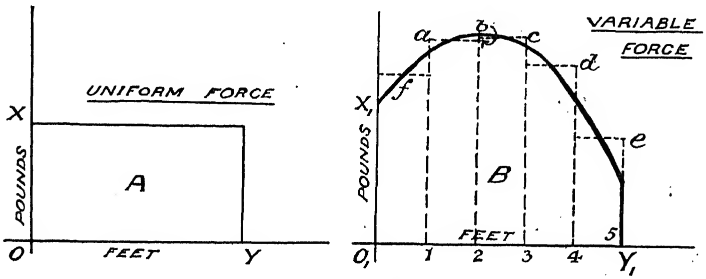
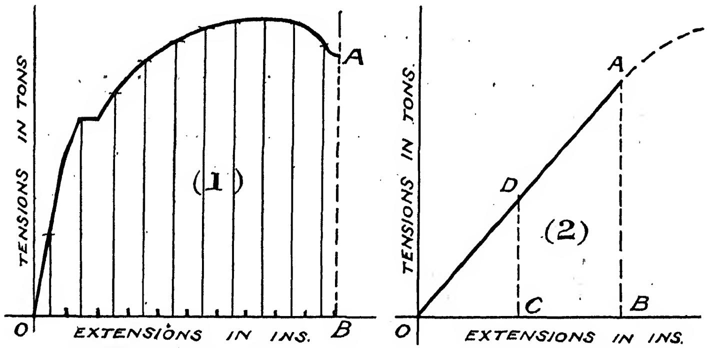


Diagram of Work, with uniform or variable forces. *Fig 325*



(Total Work) *(Resilience)*

Work done in deforming a bar.

Fig. 326.

$$\text{Work done} = \text{area } AOB = CD \times OB,$$

or generally,

$$\left.\begin{array}{c}\text{Any work within}\\ \text{elastic limit}\end{array}\right\} = \frac{\text{final max. tot. stress}}{2} \times \text{tot. strain} = \frac{F}{2} \times \Delta \text{ (inch [tons)}$$

Stress caused by Impulsive Load.—When a body moves with a given velocity, its store of energy (or work capacity) $= \dfrac{wv^2}{2g}$ ft. lbs. (see p. 98). If this be absorbed by an elastic material, we have:

$$\text{work stored} = \text{work given out}$$

$$\therefore \quad \frac{wv^2}{2g} = \frac{F^{\text{lbs}}}{2} \times \Delta^{\text{ft}} \text{ (within elastic limit)}$$

$$= \text{total mean stress}^{\text{lbs}} \times \Delta^{\text{ft}} \text{ (for all cases)}$$

and

$$\underline{\text{Total mean stress in lbs.}} = \underline{\frac{wv^2}{2g\Delta^{\text{ft}}}}$$

which is applicable to steam-hammers, pile-drivers, fly-presses, gun-targets, &c.

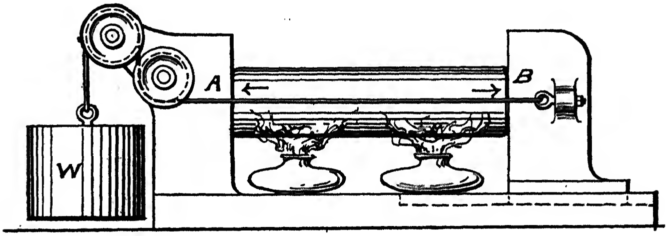
If the fall of a weight deflect a beam, or stretch a crane chain then—

$$\left.\begin{matrix}\text{work stored in weight}\\ \text{in inch lbs.}\end{matrix}\right\} = \left\{\begin{matrix}\text{work done on material}\\ \text{in inch lbs.}\end{matrix}\right.$$

$$w\,(h+\Delta) = \frac{F^{\text{lbs}}}{2} \times \Delta$$

and F^{lbs} is the greatest total stress, or the steady load which would produce the same strain Δ.

Stress caused by Heating and Cooling.—Experiment shews that the expansion or contraction by heat or cold of a bar



Force caused by heat. *Fig. 327.*

of given material, is a regular quantity for each degree of temperature. When measured per inch length or breadth, and per degree Fahrenheit, it is given in the following table :—

COEFFICIENT OF LINEAR EXPANSION IN DEG. F (α).

Material.	α.
Strong steel	·00000 63
" " tempered	·00000 73
Mild steel	·00000 57
Wrought iron...	·00000 66
Cast iron	·00000 62
Brass	·0000 105
Copper	·00000 95
Bronze	·0000 111
Invar	·000000 87

If t° = rise or fall of temperature, $\alpha\, t^\circ$ = expansion or contraction for every inch, and

Each inch is increased or decreased by $\alpha\, t^\circ$ ins.

But strain by mechanical means is $\delta = \dfrac{f^{\text{lbs}}}{\text{E}}$ (*See p.* 363.)

$$\text{Then if} \quad \alpha\, t^\circ = \frac{f^{\text{lbs}}}{\text{E}}$$

$$f^{\text{lbs}} = \text{E}\, \alpha\, t^\circ$$

and total force of expansion on walls, as in Fig. 327 at A B, is

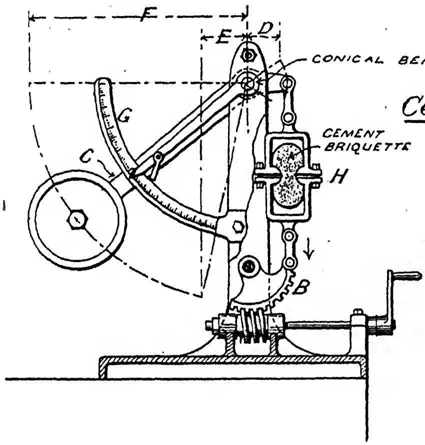
$$\underline{\text{F}^{\text{lbs}} = \text{E}\, \alpha\, t\, a}$$

Necessity of Testing to obtain Unit-strength Constants.—It has been hoped that the cohesive strength of the various materials might be obtained solely by chemical analysis, but continued experience seems to shew more and more the necessity for direct mechanical tests to obtain the strength per square inch in tension, compression, and shear; hence the use of testing machines. Certainly it is wise also to refer to chemical composition in stating the quality of a material, in order to know how far it is safe to heat or otherwise treat the same.

Testing Machines.—One machine generally serves for tension, compression, and bending experiments, the pulling shackles being changed to suit. No doubt machines will ultimately be designed to test all combined stresses, and thus verify the theoretical formulæ on which we at present rely. In small machines

the pull is exerted by turning a screw directly or by gear, but in large machines hydraulic or other power is employed, while the load is always measured by a smaller weight attached to a lever or system of levers, in steelyard fashion. (*See pp.* 834 *and* 1065.)

Cement Testing Machine.—Michele's machine will illustrate the above details, the load being applied by worm gear at B to the specimen H, a cement briquette, and the pull measured by the weight and lever C, or Danish steelyard. The arm D varies



Cement-testing Machine

Fig. 328.

very little, but the arm E increases to the maximum F or some shorter distance, during the experiment; the stress therefore varies as this arm and the pointer is left at its furthest position after rupture, while the weight returns about half an inch. The scale is graduated to represent the full load upon H.

Horizontal and vertical testing machines are so named from the direction of the pull, and each has its particular advantage; the former is represented by

The Werder Machine, extensively adopted in Germany, and shewn in Fig. 329. C is the specimen to be tested, and B an adjustable washer between shackle and crosshead A, to allow for length of C. Ram D moves to the right by water pressure from hand pumps, and the pull is given through the bolts E E, for tension at C, or compression at G. The load is measured by the

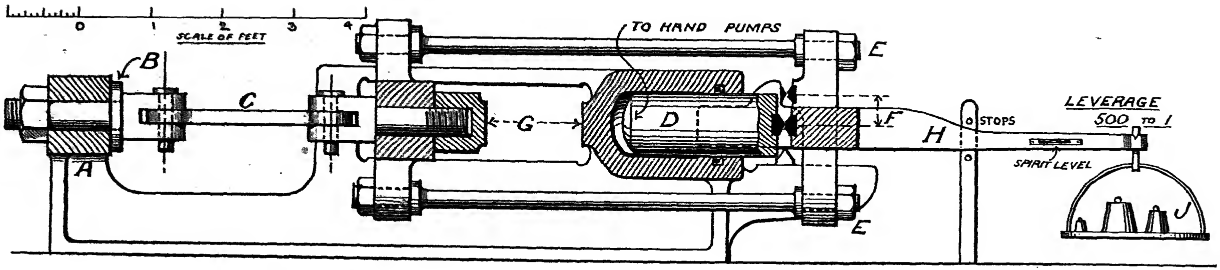


Fig. 329 Werder Testing Machine.

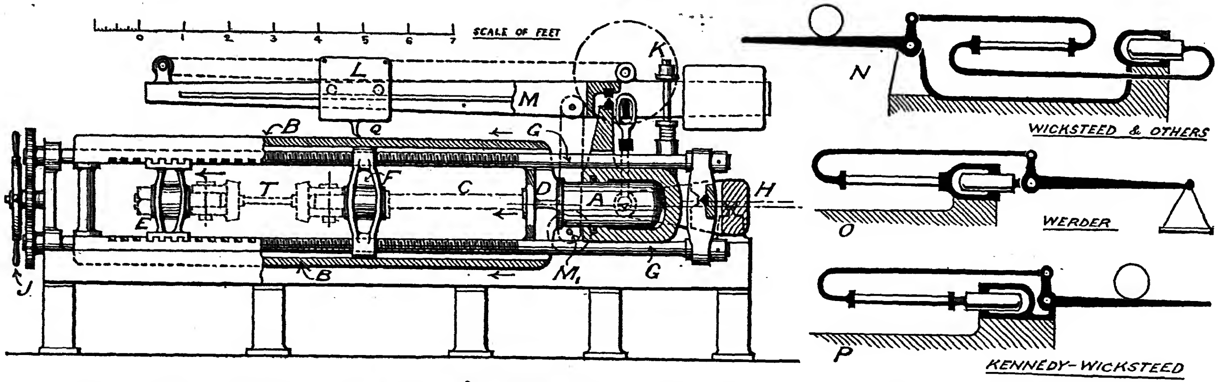
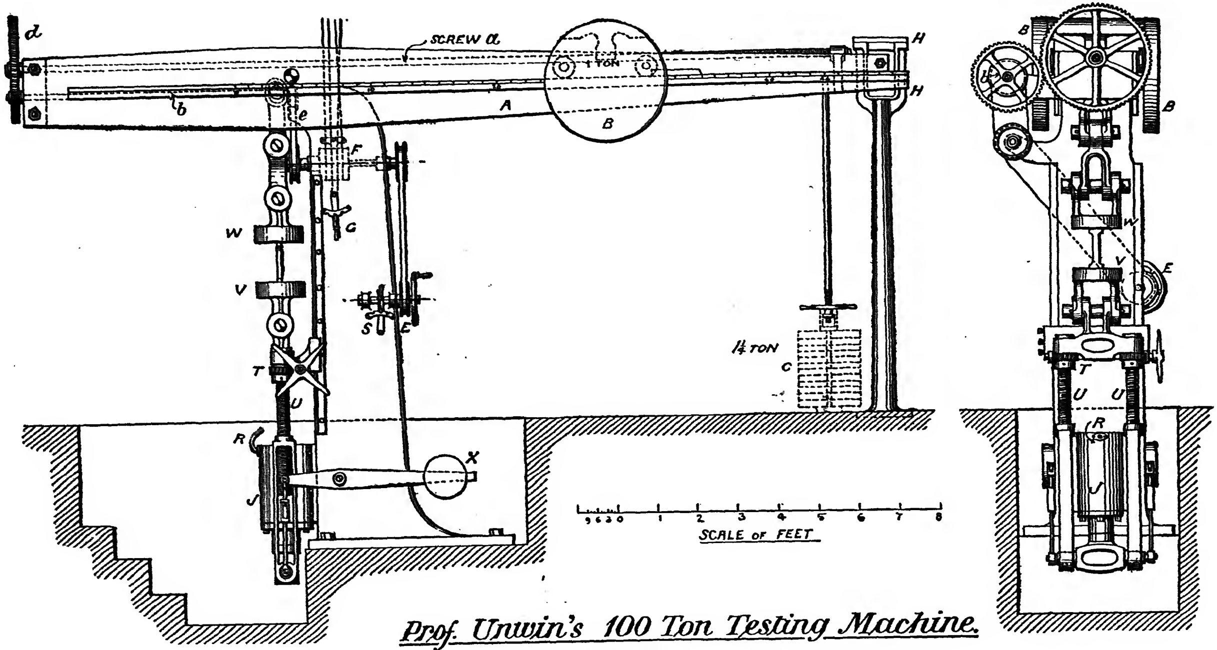


Fig. 330. Prof. Kennedy's 50 Ton Testing Machine. (BY BUCKTON & Co)

weights J and lever H, the shorter arm of which is F, the pressure being received on knife edges $\frac{3}{16}''$ apart (or much smaller than shewn), and a leverage of 500 to 1 thus obtained. A spirit level is used to ascertain the horizontality of the lever H.

Professor Kennedy's Machine.—Messrs. Buckton & Co. have made a machine to Professor Kennedy's requirements, embodying the Werder principle with improvements. In Fig. 330, A is the hydraulic ram in a fixed cylinder, and B a sliding frame carrying an adjustable crosshead E. T shews a tension experiment and C a compression experiment, the load being resisted in either case by the crosshead F, and its effect transmitted through the rods G G to the system of levers. H corresponds to H in Fig. 329, but a second lever M is here applied, with a jockey weight L to avoid the trouble of changing weights. L is traversed by hand gear at M_1 and carries a pointer at Q, while K is a spring stop, and J a hand gear for adjusting the position of F by turning the screws G G. In this machine all the operations are within control of one experimenter and the specimen well in view; in addition there is, during compression experiments, a shorter length of parts between cylinder and weighing levers than in any other machine (except the 'Emery'), as shewn by the thick lines in the figures N, O, and P, thus giving less recoil on the knife edges at rupture.

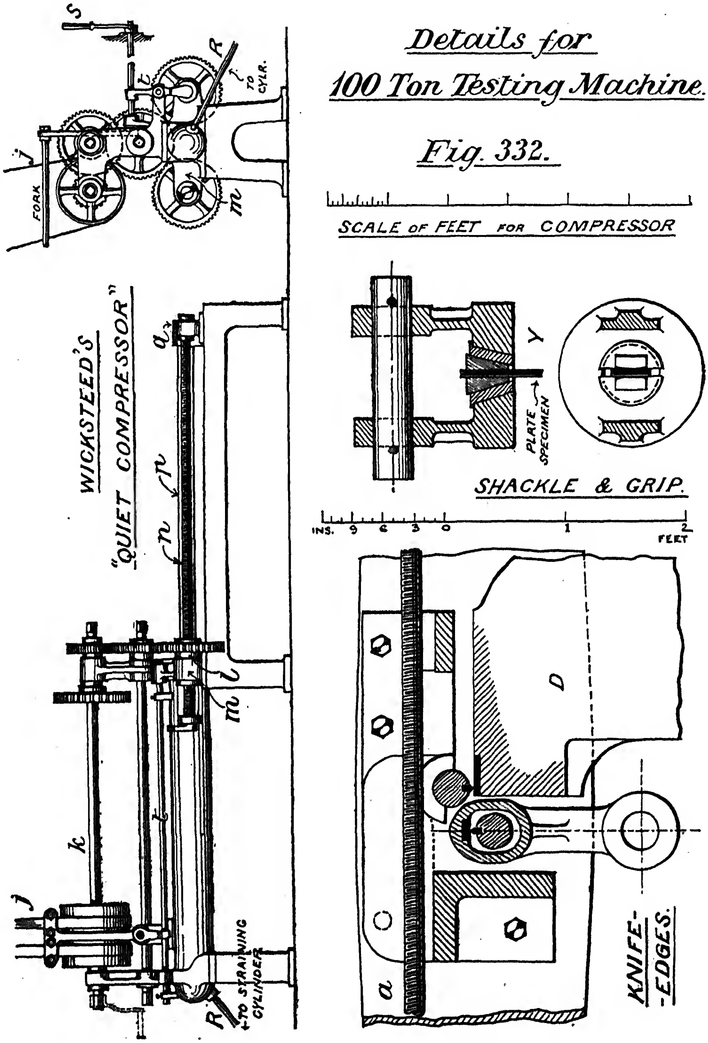
The Wicksteed Machine, also by Messrs. Buckton, is shewn at Fig. 331, as designed by Mr. Wicksteed to Professor Unwin's instructions. A is the steelyard weighing lever, and B the jockey weight, which at a leverage of 50 to 1 exerts 50 tons pull upon the specimen. Additional weights up to $1\frac{1}{4}$ tons at C exert another 50 tons by means of 40 to 1 leverage. The knife edges are shewn in detail at D, Fig. 332, being 20 inches long from front to back; and the weight B is moved by screw *a*, either by hand at E or by power at F, through the shaft *b* and gearing *d*, the connection of the strap *e* being made immediately below the fulcrum. The lever is kept horizontally between stops H H by admitting pressure water to the straining cylinder J through pipe R, and the load is relieved towards the close of an experiment by running back the jockey weight. The pressure water is obtained in Professor Kennedy's machine from the Hydraulic Power Company, in a later-built Wicksteed machine at the Armstrong



Prof. Unwin's 100 Ton Testing Machine.

(BY BUCKTON & Co.)

Fig. 331.



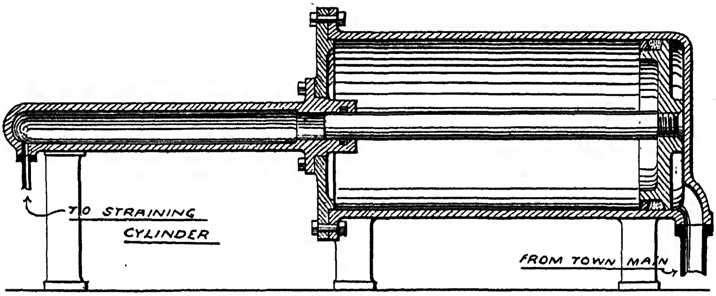
Details for 100 Ton Testing Machine.

Fig. 332.

SCALE OF FEET FOR COMPRESSOR

College by town's water acting through the Intensifier in Fig. 333, and in the usual Wicksteed machine by means of the 'Quiet Compressor' in Fig. 332. Crossed or open straps at j drive a

shaft *k*, connected by spur gear with nuts *l l*, which turn within the bosses *m m*, and thus advance the screws *n n*. The latter are connected to the ram *p* by crosshead *q*, and thus a very even pressure is given to the water, which finally passes to the straining cylinder J, Fig. 331, through pipe R. The pump may be worked by hand if necessary, or the strap fork moved by hand lever S if power be used, and a cut-off gear at *t* puts both straps on loose pulley when either end of the stroke is reached. The shackles W and V, Fig. 331, are adjusted to suit the specimen by turning the screws U U through the worm gear T; and X is to balance the

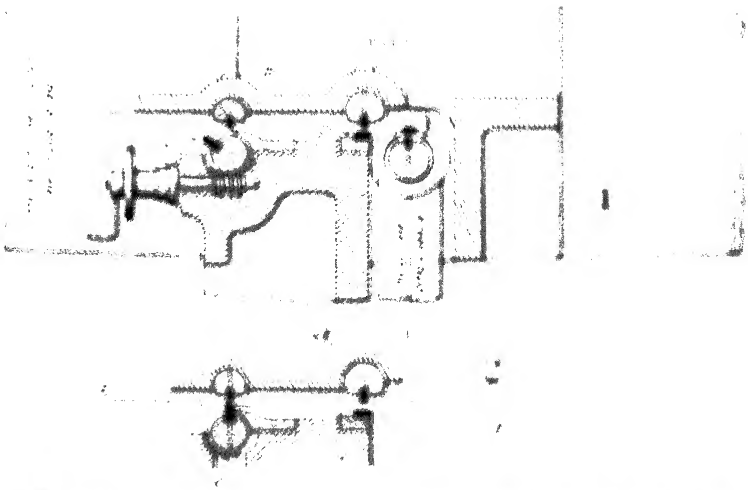


Intensifying Compressor. Fig. 333.

loose gear, from V downwards. Enlarged views of the shackles are given at Y, Fig. 332, to clearly shew the gripping wedges, slightly convex on the inside and roughed like a file.

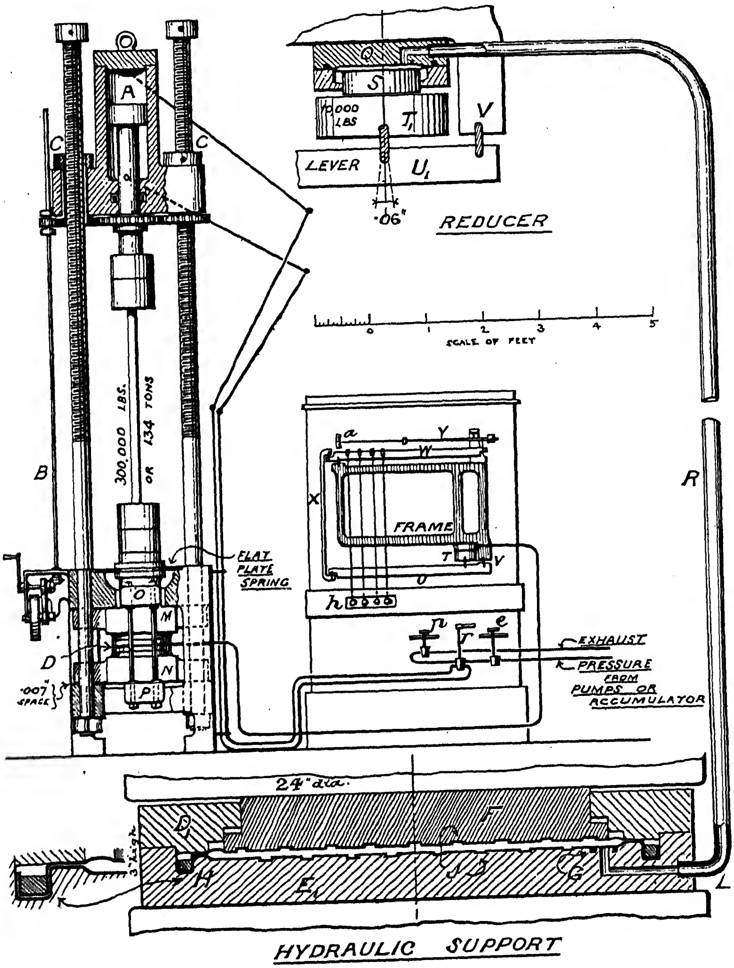
Mr. Wicksteed's alternative fulcra, as designed for Professor Hele-Shaw, are shewn in Fig. 334. Fulcrum A is employed for heavy tests, and B for lighter tests, which are thus made with a greater degree of sensitiveness. The lever knife-edges are level, but the support C, which can be put in or out of position by worm-gear, is higher than support D, as seen at (2). This gives enough clearance for vibration either at (1) or (2), and the lever takes the position E F when changing the centres. (*See App. II., p.* 836.)

The Emery Machine has obtained great favour as an instrument of precision. Professor Unwin says of a 75-ton machine: 'Every half-pound of load was precisely and instantly measured, whatever the stress the machine was exerting.' It is



Wicksteed's Hydraulic Centres Fig. 311

through hand gear X, to move the cylinder, and the water pipes have swivel joints to allow the motion. To measure the load the hydraulic support at H is employed, which consists (see enlarged sections) of a sealed sac of soft sheet brass J, .005" thick, containing alcohol and glycerine, and supported by a dwarf piston r, and cylinder c. The pressure compels the plates to fill concentric channels at J J, while further support and closure is given by the rings at M. The 'yokes' W and X take the hydraulic support between them, and the crossheads O and P in turn act on the yokes, the first for compression, and the second for tension. Thus the load, being applied on the straining cylinder, is felt at J, and transmitted to the liquid, through pipe R, to the second sac at Q.



The Emery Testing Machine. Fig. 335.

termed the 'reducer,' and from thence to the lever weighing apparatus. The movement of F is only ·001", but the reducer and support areas being as 1 : 30, the movement of piston S is

[illegible]

[illegible] Thereafter [illegible] weights [illegible] are used [illegible] Very small weights [illegible] that are necessary [illegible] and these are placed [illegible] the paper [illegible] [illegible]

The pistons [illegible] are balanced by means of springs which can be very finely adjusted, and [illegible] the [illegible] ance reduced to a very small amount [illegible] light springs through infinitesimal distances, but these are allowed for in the calibration of the weighing apparatus. Two hand wheels are placed at *f* and at *e*, the larger opening the pressure and exhaust respectively, while the smaller wheels adjust the openings within the larger plugs, so that the flow may be regulated between a foot per min and one [illegible] per hour, and *c* is a reversing valve for changing flow to top or bottom of cylinder. The Watertown arsenal machine, U.S.A., is an Emery machine of [illegible] tons capacity.

Professor Thurston's Torsional Testing Machine is designed for twisting stresses only, and was first made about [illegible]. Fig. 356 shows its construction; *a* is the specimen, held in shackles, the twist being applied by worm gear *b*, turned by hand. The load is measured by a pendulum weight *c*, as in Wicksteed's machine, and a diagram obtained autographically, in a manner to be described later.

Shackles for Test Specimens should be carefully designed to give a perfectly axial strain, or the amount of ultimate load will be affected. The Wicksteed tension shackles have already been described at Fig. 35[illegible]. Of other tension shackles in Fig. 357, *a* is Professor Unwin's, having spherical seats screwed to the ends of the specimen, a very good plan; and *b* was used by Sir John Anderson, with split sockets *c* to connect the shackles

to the specimen *b*, held in place by a slightly conical ring *c*. Compression shackles are shewn at Fig. 338. A_1 and B_1 are to take small specimens in a tension machine, and the arrangement at C_1

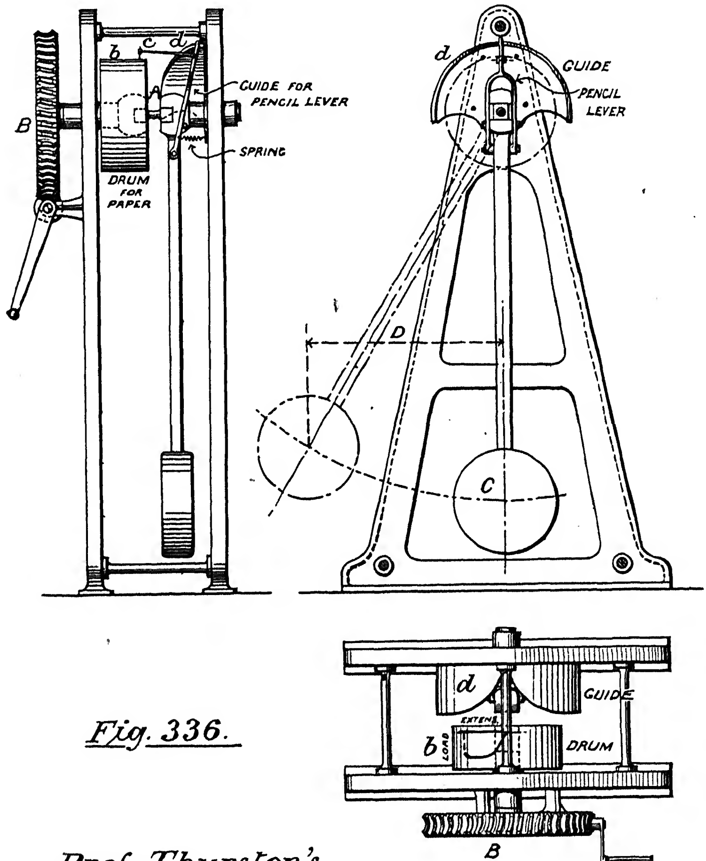
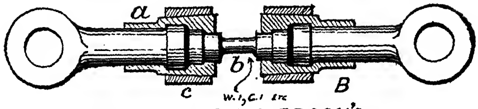


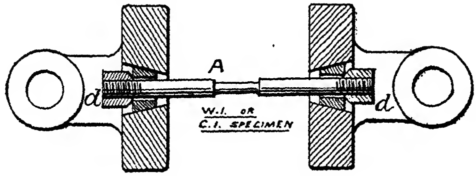
Fig. 336.

Prof. Thurston's Torsional Testing Machine.

is for admission of large specimens in a compression machine. The specimen at A_1 is placed at *c*, and the plunger *d* guided within a cylinder. As one shackle *a* slides within the other

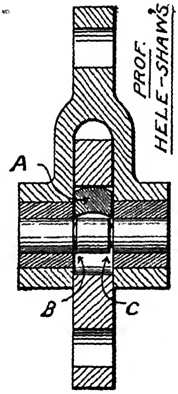


SIR JOHN ANDERSON'S



PROF. UNWIN'S

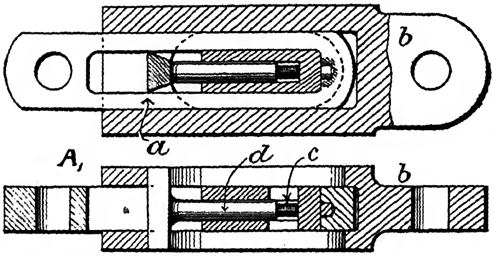
Fig. 337. Tension Shackles.



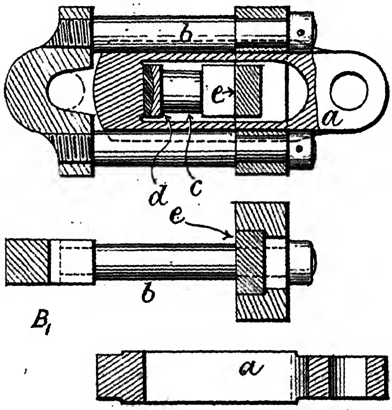
PROF. HELE-SHAW'S

Shearing Shackles.

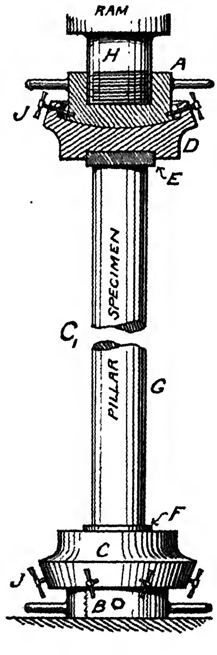
Fig. 339.



SIR JOHN ANDERSON'S



PROF. UNWIN'S



EMERY'S

Compression Shackles.

Fig. 338.

shackle *b*, a very good axial thrust is obtained. Professor Unwin's shackles at B_1 receive the test piece between a hard block *e*, and spherical surfaces *d*, and the parts are shewn separately to make their construction clear. The Emery machine is provided, for compression, with spherical nuts A and B, upon which lie convex plates or tables D and C, and the hard seatings E F receive the thrust. C and D are adjusted to the specimen by means of the handles J J. In the shearing shackles at Fig. 339 (designed by Mr. Wicksteed for Professor Hele-Shaw), a knife A adjusts itself so as to give equal pressure at B and C, while the specimen is nicked down to localise the strain. The torsion grips at A, Fig. 336, have sockets to receive a square bar turned down in the mid portion; and Fig. 340 illustrates a pair of bending shackles where knife edges B B are adjustable for various lengths of specimen, and the shackle A is formed so as to indent the bar as little as possible.

Strain Measuring.—At first it was considered sufficient to know the breaking load in tension, then Mr. Hodgkinson shewed the necessity for compression tests, and Mr. Kirkaldy lastly pointed out that the contraction of area at fracture was not to be overlooked. Now it is considered imperative to know the breaking load and elongation (usually given per cent., or extension × 100), and advisable to obtain both load and extension within the elastic limit. A stress-strain diagram, as in Fig. 321, will shew the whole life of the bar, and can be obtained in two ways: (1) by noting load and extension at several points during the experiment (the latter being measured by instruments of more or less precision), then plotting a diagram to these dimensions; or (2) by compelling the machine to make an autographic diagram.

Taking (1), the simplest method is to make a centre pop near each end of the specimen, and measure the distance between these by means of dividers; a better result is obtained by the use of a standard rod C (Fig. 341), and wedge gauge D, placed between clamps A and B on the specimen; and very great accuracy by the aid of an extensometer. Such an instrument is absolutely necessary for the fine extensions within the elastic limit, and Fig. 342 shews a very effective form devised by Professor Unwin. A is the specimen to which Tee brackets C and D are clamped, both of which carry spirit levels F and J, while D in addition supports the measuring pillar G. Within G is a fine screw carrying a

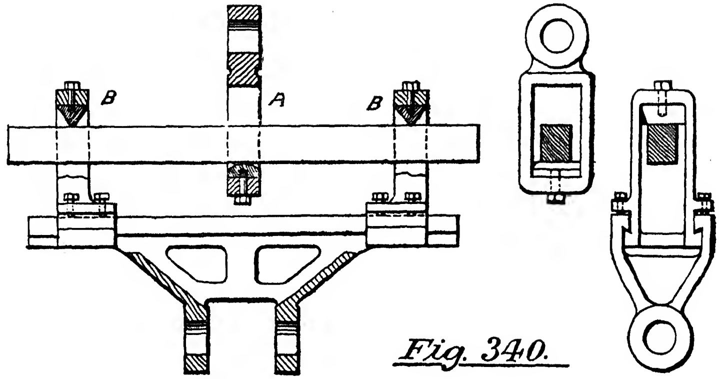
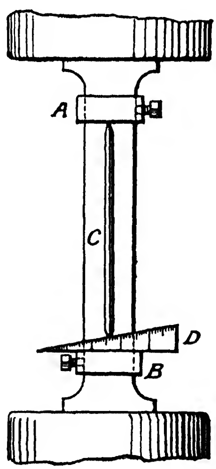


Fig. 340.

Bending Shackles (FOR WICKSTEED MACHINES)



Wedge Gauge

Fig. 341.

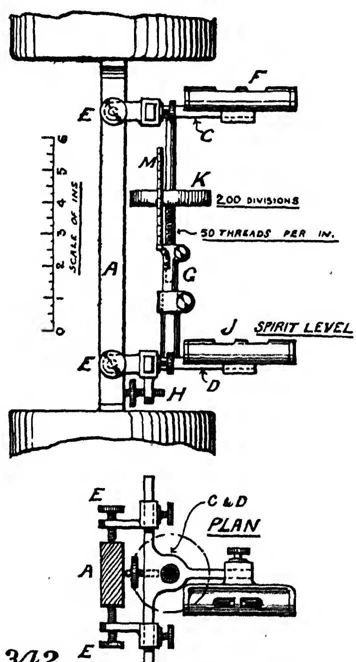
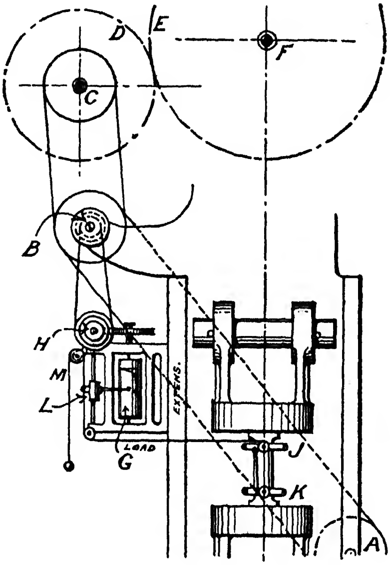


Fig. 342.

Prof. Unwin's Extensometer.

PROF. UNWIN'S

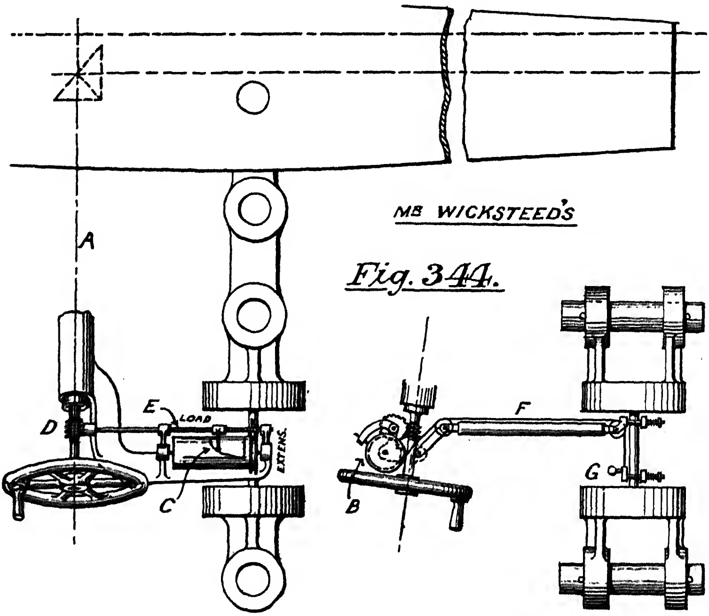
Fig. 343.



Autographic Stress-Strain Recording Apparatus

MR WICKSTEED'S

Fig. 344.



[illegible]

Autographic Test Diagrams [illegible] Among the earliest were those of Professor Thurston [illegible].* Paper is carried on the drum [illegible] while a pencil attached [illegible] proportional to the arm [illegible] in a direction at right angles to the plane of the latter, thus representing the load, and the extension, measured in degrees, is given by the rotation of the drum itself relatively to the pendulum.

Although Mr Wicksteed has obtained good diagrams by using the pressure water to represent the load, the method is [illegible] considered advisable, on account of the [illegible]. Better results are obtained by the apparatus [illegible]. The first will be understood by comparing with Fig. [illegible]. The rotation of the [illegible] weight screw being proportional to the load, it follows that the screw-turning gear can be used to rotate a paper drum [illegible] and thus represent [illegible] a pencil [illegible] may be moved vertically by a wire from the specimen, to show the extension; [illegible] are some pulleys which drive the drum through worm gear, and at the same time allow of change of [illegible] scale; and the wire is wrapped upon itself before leaving the slope [illegible], so as to mark the extension to twice the natural scale. [illegible] is the hand wheel connected to the countershaft [illegible] by belting, and [illegible] is the auxiliary shaft, which turns screw [illegible] through spur gear.

To avoid errors of belt slip between [illegible] and [illegible] the gear in Fig. [illegible] has been introduced. Here the screw is turned by motor gear through shaft [illegible], and a pencil [illegible] represents the load, being traversed by screw at [illegible] turned by worm gear [illegible]. At the same time the drum [illegible] is rotated for extension by a wire [illegible] from the specimen, and a hammer weight [illegible] releases the wire when rupture

* The writer's attention has been called to Mr Cowley's apparatus [illegible], which probably antedated the above. Fig. [illegible] was obtained from a

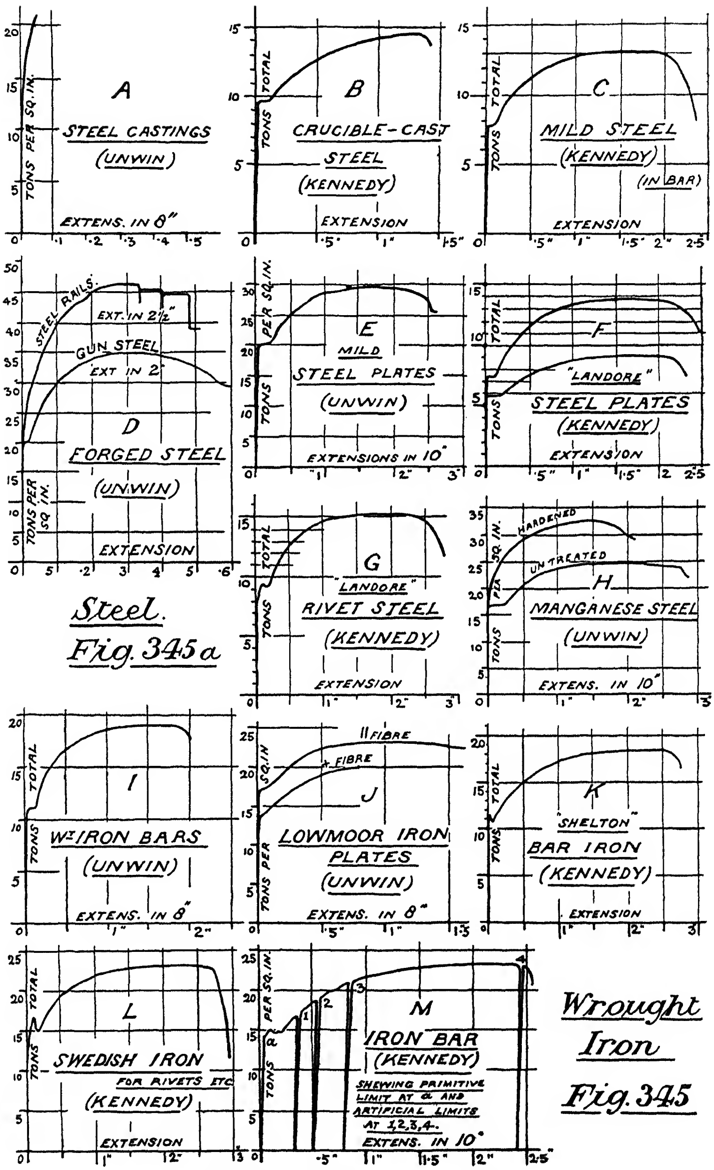
occurs. This apparatus has been applied in Professor Hele-Shaw's machine.

Stress-strain Diagrams, as obtained principally by the previous apparatus, will now be shewn (*see* Figs. 345 *and* 346). The largest number of experiments have been made in tension and our list of compression and shear diagrams is but meagre In every case the authority has been cited, and where possible the unit stress and length of specimens given.

Deductions.—Mild steel and good wrought iron have long plastic extensions and considerable contraction at rupture (see C, F, G, L). Stronger steels are less ductile, as at B and D, while steel castings, A, are very short, though the strength may be higher than shewn. Cast iron, Q, has really no elastic stage, though Hodgkinson fixed an apparent limit, but brass, O, is better off, and is much more plastic. N is a very fine diagram for aluminium bronze, shewing great ductility and high elastic limit. Torsional and transverse diagrams (S and R) are not essentially different from tension in character, but compression diagrams take quite a different form, V being a typical example, the plastic portion tending always to curve in an opposite direction to that of tension. T is an experiment on long pillars held loosely in sockets to prevent bending; and diagrams Q, T, U, and V have all been plotted.

Raising the Elastic Limit.—If the load be carried a little beyond the primitive elastic limit and allowed to remain, say, for 24 hours, then removed, the bar will strain slightly; but on re-stressing, a new elastic limit will be found at a little higher load than that just removed. Repeating the experiment beyond the second limit, a third limit may be found, and so on until the bar breaks. All this is beautifully given by diagram M, and also by diagram S, one plastic curve bounding all the limits, and it is clearly shewn why English engineers consider the breaking load the only reliable test of a material. (*See Appendices, pp.* 756 837, 1071, *and* 1074.)

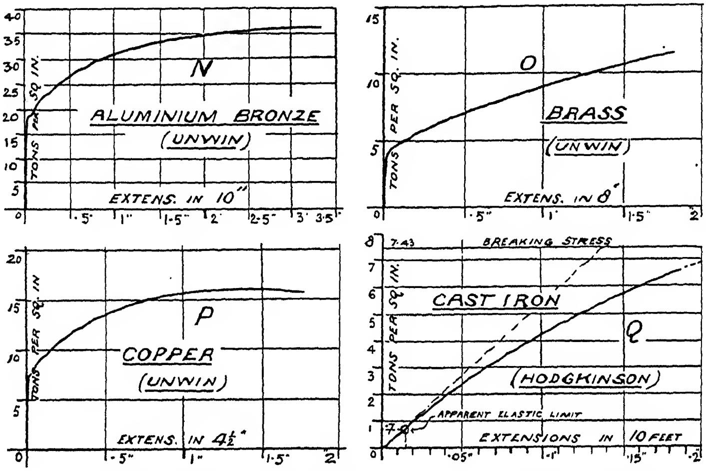
Local Extension.—In Fig. 347 a test strip has been taken 12″ long, and divisions marked across it at one inch apart, then the actual extensions within each inch measured, and set up as ordinates on the line A B; C D E is the curve shewing distribution of extension, and is seen to increase very greatly towards the fifth



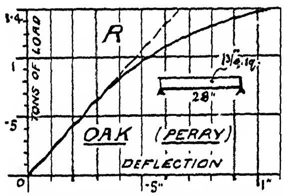
Steel. Fig. 345a

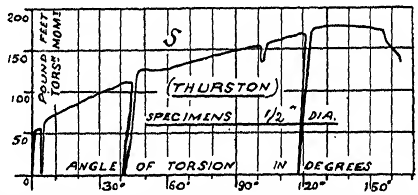
Wrought Iron Fig. 345

Stress-Strain Diagrams.

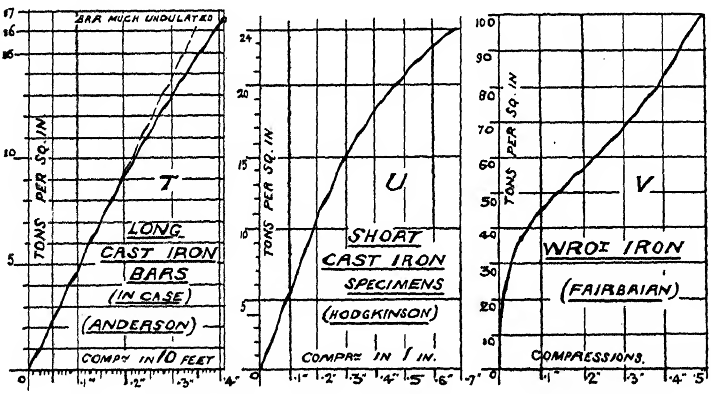


Tension Diagrams Fig. 346a.





Bending Diagᵐ Fig. 346b. Torsion Diag. Fig. 346c.



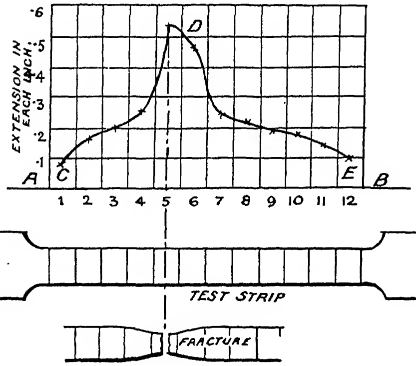
Compression Diagrams. Fig. 346.

inch, where fraction occurred. This indicates the necessity of stating elongations somewhat as follows:—'28·2 per cent. in a length of 8″,' or '25·8 per cent. in a length of 10″,' meaning ·282 or ·258 of the original length; and the breaking stress should be measured as maximum load ÷ original area. (*See p.* 837).

Diagrams shewing the elastic line have also been drawn by Mr. Thos. Gray, of America, by means of the double apparatus shown at Fig. 348. The paper drum is rotated by worm gear, as in Fig. 343, to give the load, and there are two pencils H and C, both connected to the specimen by wires; but while A is connected to C through the single lever B and gives an ordinary diagram, D gives motion to H through the triple set of levers E, F and G, and thus the stroke of H is very greatly magnified. Three diagrams are shewn, where the higher curves are drawn by C, and the lower or elastic lines by H. Of course two extension scales are required.

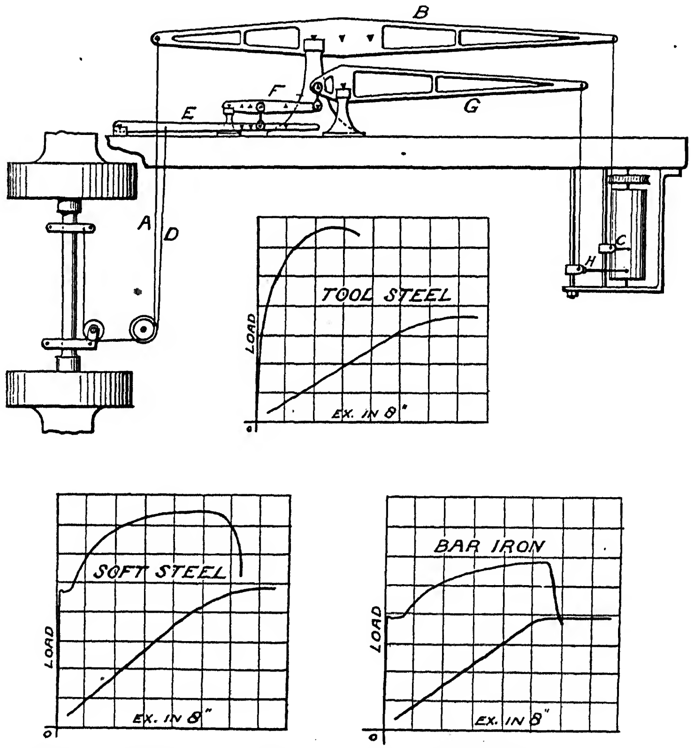
Admiralty Tests.—All war material must be tested as follows, the data serving also as a general standard :—

	Material.	Tensile breaking stress in tons per sq. in. of original area.	Elongation.
	W. I. Ship Plates (1st class) ...	22 ‖ / 18 +	...
	W. I. Ship Plates (2nd ,,) ...	20 ‖ / 17 +	...
	W. I. Section Bars	22	...
	W. I. Boiler Plates	21 ‖ / 18 +	...
	Steel Ship Plates	26 to 30	20 % in 8″
For Gun Mountings.	,, Castings (intricate)	28 minimum	13½ % in 2″
For Gun Mountings.	,, ,, (Roller Paths and Pivot Plates)	36 to 40 : yield point at 18 min.	13½ % in 2″
For Gun Mountings.	,, ,, (Girders, Cylinders, and 'Ordinary')	28 minimum	18½ % in 2″
For Gun Mountings.	Steel Rivets	27 maximum	...
For Gun Mountings.	,, Forgings (general)	28 to 35	28 to 24 % in 2″
For Gun Mountings.	,, ,, (Piston Rods) ...	32 to 35	28 to 24 % in 2″
For Gun Mountings.	,, ,, (Rollers and Roller Paths)	38 to 45	22 to 16 % in 2″
For Gun Mountings.	,, Plates	28 to 32	20 % in 8″
For Gun Mountings.	Gun Metal (ordinary)	14 minimum	7½ % in 2″
For Gun Mountings.	,, (for hydraulics) ...	,,	3½ % in 2″
For Gun Mountings.	Manganese Bronze	28	25 % in 4″



Distribution of Extension

Fig. 347.

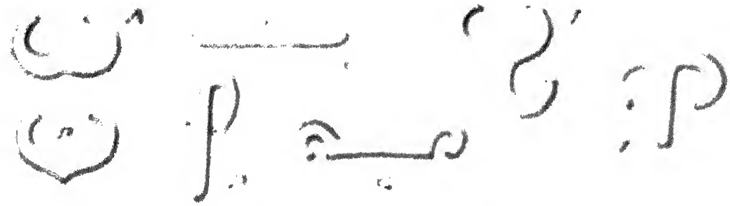


Double Test Diagrams & Apparatus. Fig. 348.

[illegible]

[illegible] Mild Plates.

Hot tests [illegible]



Alternating Tests on Rolled Bars. Fig. 340.

Cold tests are [illegible] bending and breaking to observe the fracture [illegible], as with wrought-plate tests, a tensile fracture [illegible] dropping a weight of one ton through [illegible] feet [illegible] bent by a falling weight [illegible] the conditions of practice. (See *Appendix II., p.* [illegible].)

[illegible] hammering while [illegible], and punching [illegible] with a tool of [illegible] without cracking in either case.

Wohler's Law. In 1870, Herr Wohler conducted experiments on variation of stress, and showed conclusively that the range of variation was a factor in lowering the breaking load, so that a bar under variable stress would break much more easily than if an unchangeable load were applied. His experiments were so conducted that the test bars withstood two or three million changes of load before breaking. Prof. Unwin, who has given great attention to endurance tests, deduces the following general equation:

$$f_2 = \frac{\delta}{2} + \sqrt{f_1^2 - n\delta f_1}$$

where f_1 = original breaking stress in tons sq. in.

δ = stress variation in terms of f_2 in tons sq. in.

n = a constant deduced from experiment

$= \left(\begin{array}{l} 1.5 \text{ for Wrought Iron and Mild Steel} \\ 2.0 \text{ for Hard Steel} \end{array} \right)$

f_2 = new breaking stress in tons per sq. in.

Taking $s = 1.5$, three simple cases may be deduced:

(1.) A steady or dead load

$$\underline{f_2 = f_1}$$

(2.) A simple live load, or 'suddenly applied load,' viz., one removed and replaced continually and instantly, but without velocity

$$S = f_2 - 0$$

and $$f_2 = \frac{f_2}{2} + \sqrt{f_1^2 - 1.5 f_2 f_1}$$

Simplifying, squaring, and solving the quadratic obtained:

$$\underline{f_2 = .6 f_1}$$

(3.) Reversal of stress, or alternate compression and tension of equal value, as in rotating shafts

$$S \text{ is from } + f_2 \text{ to } + f_2 = 2 f_2$$

and $$f_2 = \frac{2 f_2}{2} + \sqrt{f_1^2 - 1.5 \times 2 f_2 f_1}$$

$$\underline{f_2 = \tfrac{1}{3} f_1}$$

Summing up, we have for

Steady load . . $f_2 = f_1$
Live load . . . $f_2 = .6 f_1$ for Wrought Iron; $.8 f_1$ for Steel
Reversible load $f_2 = \frac{1}{3} f_1$ for Wrought Iron; $\frac{3}{8} f_1$ for Steel

or roughly, the strengths are as 3 : 2 : 1. (*See Appendices, pp.* 756, 839, *and* 1071.)

Factor of Safety.—The working stress must not only be within the primitive elastic limit of the stress diagram, it must also be further reduced on account of stress variation, and still further because the working conditions can rarely be all estimated; the correction for all these being made as follows:

$$\text{Load or unit stress (safe)} = \frac{\text{load or unit stress (breaking)}}{\text{factor of safety}}$$

If a foundation factor of 3 be used to cover uncalculated effects, and to keep within the elastic limit, then, by Wöhler:

A steady load requires a factor . . .	3	Wrought Iron	2	Steel
A live load requires a factor	$3 \times 2 = 6$		4	
A reversible load requires a factor	$3 \times 3 = 9$		6	

And the following table, deduced from practice, is fairly explained:

FACTORS OF SAFETY.

Material.	Dead Load.	Live Load.	Moving Load.
Wrought Iron and Mild Steel	3	5 to 8	9 to 13
Hard Steel	3	5 to 8	10 to 15
Bronzes	5	6 to 9	10 to 15
Cast Iron and Brass	4	6 to 10	10 to 15

Average Stresses adopted in practice.—We must now sum up the results obtained in testing, as given by the best authorities, and form a table of breaking and safe stresses. But as there are high and low qualities for each material, and samples of each quality vary so much, our tabulations can only be the averages of many averages.

Breaking Stresses.—Thus *cast iron* may vary from 5 to 15 tons per square inch in tension, 22 to 58 in compression, and 4 to 5 in shear. *Wrought iron* breaks at from 15 to 30 tons in tension, and 10 to 22 tons in shear. The strength of *steel* increases with the carbon it contains, but as a rule its elongation is simultaneously decreased. *Steel plates* should have but $\frac{1}{4}$ per cent. *Cementation steel* reaches very high strengths, varying from 40 to 67 tons per square inch in tension, some samples of tool steel yielding 88 tons; and tempering increases its strength. *Steel castings* bear from 15 to 34 tons with reasonable elongation. *Copper* depends on mechanical treatment. Cast, it supports 10 tons; rolled into plates, 14 tons; and drawn into wire, 20 tons. *Brass* has 8 to 13 tons per square inch tension, and *gun metal* 10 to 23 tons.

There is some difficulty in collecting good results for compression. If the specimen be ductile it flattens out, and then, as Rennie said, 'the resistance becomes enormous.' Brittle materials are more easily dealt with. Besides, tension has been looked upon as a sufficient test for all materials, and thus the compression and shear columns are in many cases vacant. In such cases we may take compression = tension, and shear = ·7 of tension.

The safe stresses given in the table are those usually adopted in English practice, and have a factor of 5 or 6 on the breaking stress. Continental engineers take the elastic strength as their guide, but its unstable character prevents its adoption as the standard in this country. In deciding upon the working stress, the designer should, however, consider well the following four heads:

1. *Elastic limit.* Some idea of its position should be obtained. Compare 1 and 9, Figs. 145 and 146, a higher proportionate stress being allowable in the former.

2. *Variation of stress.* The condition of loading should be found with care.

3. *Unknown actions.* Endeavour to ascertain to what extent these occur, and whether they form an important part of the total load.

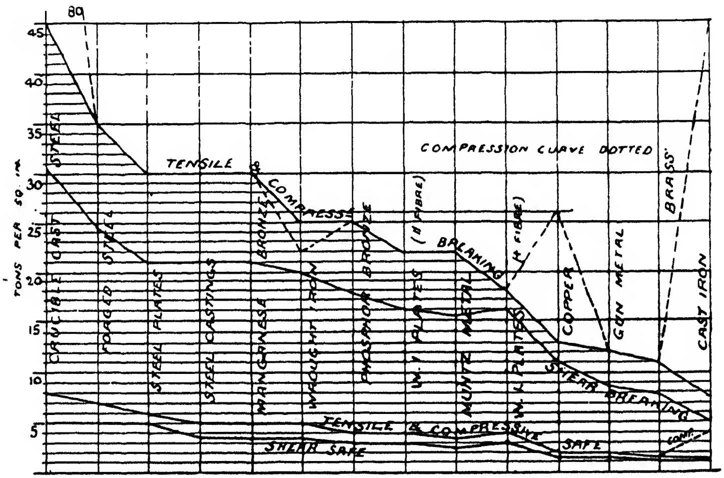
4. *Quality of material.* If possible, a test for all material should be insisted on, both for ultimate load and elongation; preferably also for contraction of area and elastic strength. This will enable the designer to fix his working strength with great exactness. (*See Appendix II., p. 818.*)

TABLE OF THE AVERAGE BREAKING STRENGTH OF MATERIALS AND SAFE STRESSES FOR ORDINARY LIVE LOADS (IN TONS PER SQ. IN.).

Material.	In Tension.		In Compression.		In Shear.	
	Breaking.	Safe.	Breaking.	Safe.	Breaking.	Safe.
Steel (Crucible cast, for strong forgings and tools)	45	8	80	8		5
Steel (mild, for general forging)	35	7		7	...	5
Steel Plates (and rivet steel)	30	6		6	24	5
Steel (for castings)	30	5	...	5	...	4½
Manganese Bronze	30	5	...	5	...	4½
Wrought Iron (forgings)	25	5	22	4	20	4½
Phosphor Bronze	25	4	...	4	...	3
Wrought Iron Plates (with grain)	22	4	...	4	16	3
" " " (across grain)	18					
Muntz Metal	22	3½		3½	...	2½
Copper	13	2	26	2	11	1½
Gun Metal	12	2	...	2	...	1½
Brass	11	1½		1½		1
Cast Iron	7½	1½	45	4	5	1

The quantities in this table are given in tons, because the numbers are thus more easily remembered, and because it is the Engineer's language. Fig. 350 shews them diagrammatically.

(*See Appendices, pp.* 840 *and* 1075 *for further materials.*)



Average Stresses for Materials *Fig. 350.*

The Proportioning of Structures and Machine Parts by Calculation.—The equality of action and reaction is the starting point in constructive calculation. Whether the load be applied directly or through a lever arm, the external forces must balance the internal stresses, and we have for the two cases:

(Direct action). Total load = Total stresses.

(Lever action). Moment of load = Moment of stresses.*

which are our general strength equations.

Classification of Stress Action. — Practical cases of simple or compound stress may be arranged under ten heads:—

* A moment = force × lever arm.

Kind of Stress.	Some Cases.
1. Tension	Lifting rods, chains, bolts, ropes, boiler shells, pipes and cylinders, boiler stays, flywheel [rims.
2. Compression	All short pillars.
3. Shear	Punching and shearing, rivets, pins, cotters, coupling bolts, keys.
4. Torsion (momental shear)	Short shafts, spiral springs.
5. Bearing	Plate edges on rivets, cotter edges, and cantilevers.
6. Bending (momental compression and tension)...	Beams, axles, boiler end plates, slide bars, teeth of wheels.
7. Bending + Tension ...	Crane hooks.
8. Bending + Compression	Long pillars, boiler flues, ships' davits, connecting rods.
9. Torsion + Bending ...	Long shafts, crank arms.
10. Torsion + Compression...	Propeller shafts.

Tension Stress-Action.—Unit stress × area of section will give total stress. Therefore :—

$$\text{Load} = \text{Total stress.}$$
$$W = f_t\, a.$$

In the case of steam or water pressure the load is unit pressure × area pressed upon, and

$$p^{\text{tons}} \times \text{area of boiler end, or piston, in square ins.} = f_t\, a.$$

Of course f_t may be either 'breaking' or 'safe,' and W or p will vary in like manner.

Example 1.—Find safe load for following sections, at 5 tons per square inch. (1) 3 ins. dia. (2) 3 ins. dia. with $\frac{3}{4}''$ cotterway. (3) $3''$ tube with $2''$ hole. (Eng. Exam., 1892.)

(1) $a = \pi r^2 \qquad = \frac{99}{14} \quad \therefore\ \underline{W = fa = 35{\cdot}35 \text{ tons.}}$

(2) $a = \pi r^2 - \frac{3}{4}d \quad = \frac{135}{28} \quad \therefore\ \underline{W = fa = 24{\cdot}10 \text{ tons.}}$

(3) $a = \pi r_1^2 - \pi r_2^2 = \frac{55}{14} \quad \therefore\ \underline{W = fa = 19{\cdot}64 \text{ tons.}}$

Example 4. A steady load of five tons produces a deflection of half an inch. Find the height a weight of [illegible] cwt. must fall to produce the same deflection. (Eng. Exam., [illegible].)

[illegible]

Example 5. Find the dia. of a W.I. pump rod, the bucket being 30 ins. dia., and the water pressure 15 lbs. per square inch. f = [illegible] tons. (Eng. Exam., [illegible].)

[illegible] area of bucket = $f \cdot a$

[illegible]

d = [illegible] ins.

The Strength of Chain.—Both sides of the link resist tension, so taking $f = 4$ tons safe:

$$W = 2 \times \pi r^2 \times 4 = 25{\cdot}12 \, r^2$$

but $r = \frac{d}{2}$ $\therefore$ $\underline{W = 6{\cdot}28 \, d^2 \text{ tons safe load.}}$

Sir Jno. Anderson deduces a simple rule from the above:

$$\frac{(\text{dia. in eighths})^2}{10} = \text{safe load in tons.}$$

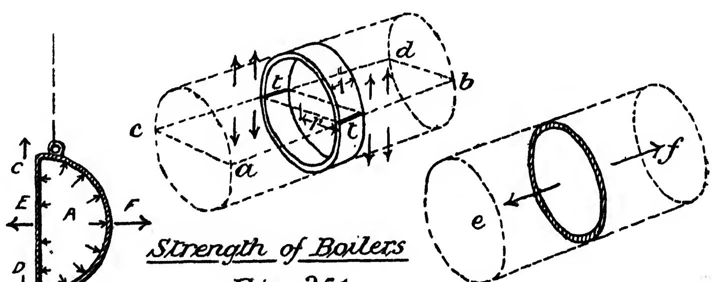
Thus an inch chain bears $\frac{8 \times 8}{10} = \underline{6{\cdot}4 \text{ tons.}}$

Strength of Ropes.—For *white hemp* $f_t = \frac{1}{2}$ ton safe. But as all ropes are measured by their circumference, and area $= \frac{\text{circ.}^2}{4\pi}$

$$\underline{\text{Strength of hemp rope} = \tfrac{1}{2} \times \frac{\text{circ.}^2}{4\pi} = {\cdot}04 \text{ circ.}^2 \text{ (tons).}}$$

Wire rope has its members stated by their W.G. Referring to page 276, the total area may be reckoned: then let $f_t = 11\frac{1}{2}$ tons safe for iron or steel.

Strength of Pipes and Cylinders, pressed internally. Imagine a hemispherical vessel A, Fig. 351, hung by a string, and



Strength of Boilers

Fig. 351.

pressed internally; then, as the vessel moves neither to right or left, it follows that the total pressure on the curved surface *in direction* F is equal to that upon the flat surface. The flat surface is called the 'projected area' of the curved surface.

First Case, Thin Cylinders.—A boiler, or thin cylinder *a b c d.* Fig. 351, tends to tear along the joints *a b* and *c d.* Examining a strip 1″ wide we obtain:

$$\left.\begin{array}{c}\text{Internal load on}\\ \text{projected area}\end{array}\right\} = \left\{\begin{array}{c}\text{safe strength of two sections of}\\ \text{plate (in tension).}\end{array}\right.$$

$$p^{\text{tons}} \times 2\,r \times 1 = f_t \times 2\,t \times 1.$$

$$\therefore\ \underline{p^{\text{tons}} r = f_t\, t.}$$

Suppose the plate tends to tear at a ring section as at *e f*, then

$$p^{\text{tons}} \times \pi\, r^2 = f_t \times 2\,\pi\, r \times t.$$

$$\therefore\ \underline{p^{\text{tons}}\, r = 2 f_t\, t.}$$

From this we find that

$$\text{Stress on longitudinal section} = \frac{p\,r}{t}$$

$$\text{and stress on transverse section} = \tfrac{1}{2}\frac{p\,r}{t}$$

so there is no fear of a boiler bursting at a cross seam. The above supposes the boiler plate to be of uniform construction throughout. But as the seams, whether welded or riveted, are much weaker than the 'solid' plate, a multiplier (η) must be introduced on the right side of the equation to reduce the quantity and shew the strength at the joint. Then:

$$\eta = \text{efficiency} = \frac{\text{strength of joint}}{\text{strength of solid plate}}$$

or, efficiency per cent $= \eta \times 100$

$$\left.\begin{array}{ll}\text{For lap welded joints,} & \eta = \cdot 7\\ \text{For single riveted joints,} & \eta = \cdot 5\\ \text{For double riveted joints,} & \eta = \cdot 7\\ \text{For electric welded joints,} & \eta = \cdot 85\end{array}\right\}\text{roughly}$$

(*See p.* 755.)

and the formula for boiler or pipe strength becomes

$$\underline{p^{\text{tons}} r = f_t\, t\, \eta}$$

Example 6.—A copper steam pipe 12″ dia. is to resist an internal pressure of 160 lbs. per sq. in. Find its thickness, if η for the brazed joint = 80%

From above formula $\dfrac{160}{2240} \times 6 = 2 \times t \times \cdot 8 \qquad \therefore\ \underline{t = \cdot 267 \text{ ins.}}$

Example 7.—Find the bursting resistance of a cast-iron pipe $\frac{1}{2}$ in. thick and 10 ins. diameter. (Eng. Exam., 1887.)

$$p^{\text{tons}} r = ft \text{ (there being no seam)}$$

$$p^{\text{tons}} = \frac{ft}{r} = \cdot 75 \text{ ton} = 1680 \text{ lbs. per sq. in.}$$

Second Case, Thick Cylinders.—If cylinder thickness be small in comparison to diameter, the stress at the inner surface is practically the same as at the outside. But this is by no means the case with very thick cylinders. Then the following formula must be applied, devised by Lamé:

$$\frac{R''}{r} \text{ or } \frac{D}{d} = \frac{\sqrt{f_t + p^{\text{tons}}}}{\sqrt{f_t - p^{\text{tons}}}}$$

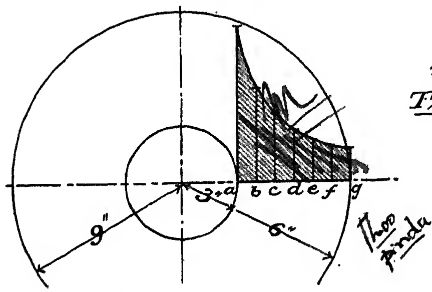
and the stress varies throughout the thickness, the hoop tension f_t being found at any intermediate radius i by the following formula:

$$f_h \text{ in tons} = \frac{p^{\text{tons}} r^2 (i^2 + R^2)}{i^2 (R^2 - r^2)}$$

Example 8.—A cast-iron hydraulic cylinder is 6″ internal diameter, and loaded with 1 ton per sq. in. pressure. Find (1) the thickness, and (2) construct a curve shewing the hoop tension throughout the thickness.

$$\frac{R''}{r} = \frac{\sqrt{f + p^{\text{tons}}}}{\sqrt{f - p^{\text{tons}}}} = \sqrt{\frac{1 \cdot 25 + 1}{1 \cdot 25 - 1}} = \sqrt{9} \quad \therefore R'' = 9 \quad \text{and } t = 6''$$

The section of the cylinder is shewn at Fig. 352, and the ordinates



Strength of Thick Cylinders.

Fig. 352.

at $a\,b\,c\,d\,e\,f\,g$ shew the hoop tension at the various rings, found as follows:

$$\text{at } a, \ f_h = \frac{1 \times 9}{9} \frac{(81 + 9)}{(81 - 9)} = 1 \cdot 25 \text{ tons}$$

Similarly at b, $f_h = \cdot 76$ ton, at $c = \cdot 53$ ton, at $d = \cdot 406$ ton, at $e = \cdot 332$ ton, at $f = \cdot 283$ ton, and at $g = \cdot 25$ ton.

The curve is an equiangular or logarithmic spiral. Large guns are built of coils shrunk one over the other, so as to put the inner tube in a state of compression. The pressure of the explosion then tends to equalise the stress, by slightly adding to the outer tension, but more than removes the inner compression. When cold, a coil is slightly smaller than the core it is to envelop, according to the following rule:

$$\text{Diminution of coil dia.} = \frac{\text{mean dia.}^2 \times c}{\text{inside dia.}}$$

where c for the outer coils $= \cdot 00133$
c for next inner coils $= \cdot 00108$
c for next inner coils $= \cdot 00083$

Let an outer coil be 17″ outside and 12″ inside, then

$$\text{Diminution} = \frac{14\cdot 5 \times 14\cdot 5 \times \cdot 00133}{12} = \cdot 0233''$$

The same effect is produced in cast-iron cylinders by casting with a cold-water core, and thus much less thickness is required. (See Figs. 289, 298, 299, 300.) (*See Appendices I. and II., pp.* 757 *and* 841.)

'Casting Rule' for Steam Cylinders, &c.—With the usual steam and gas pressures, the previous formulæ give so small a thickness that the metal would not fill the sand mould during casting, so an empirical rule must be adopted to enable the cylinder or gas pipe to be cast, thus:

$$t = \cdot 18\sqrt{d}$$

This will represent the thickness of steam chest and other parts, but the cylinder body should be about $\frac{1}{4}$ in. thicker, to allow for reboring, and the flanges should also be stiffer.

Tensile Stress induced by Centrifugal Force.—When a weight w, attached to a string, is swung in a circular path, it exerts a pull upon the string represented by the formula

$$F^{\text{lbs}} = \frac{w v^2}{g R} \text{ lbs.} \quad (\text{where } v = \text{actual velocity of weight})$$

In a grindstone or flywheel this centrifugal pull exerts a tension between the particles of the material, which we shall examine in

Fig. 353. R is the average radius (radius of gyration) of the rotating flywheel rim, $a\ h\ b$. If $w =$ the weight of a cub. inch of

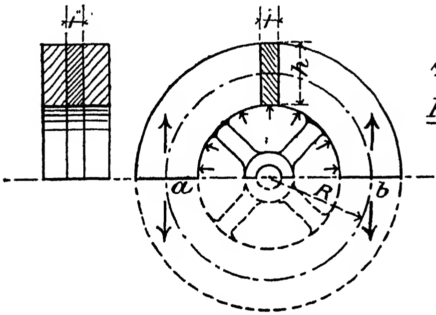


Fig. 353.

the material, $w\,h$ is the weight of the darkly shaded solid, and its centrifugal force,

$$F^{\text{lbs}} = \frac{w\,h\,v^2}{g\,R}$$

But every such solid in the circumference acts radially as at A, Fig. 351, and the flywheel tends to burst at $a\ b$, as the boiler did at $t\ t$, Fig. 351.

$$\therefore \left.\begin{array}{c}\text{Centrifugal force per}\\ \text{sq. in. of rim}\end{array}\right\} \times \text{projected area} = \left\{\begin{array}{l}\text{safe strength of}\\ \text{strip section at } a\,b\end{array}\right.$$

$$\text{or} \quad \frac{w\,h\,v^2}{g\,R} \times 2\,r = 2\,f^{\text{lbs}}\,h$$

$$\text{and} \quad f_t \text{ lbs.} = \frac{w\,h\,v^2 \times 2 \times 12\,R}{g\,R \times 2\,h} = \frac{12\,w\,v^2}{g}$$

$$\text{then} \quad v = 1{\cdot}64\sqrt{\frac{f^{\text{lbs}}}{w}}$$

For cast iron, $w = {\cdot}26$ and $f^{\text{lbs}} = 1{\cdot}25 \times 2240$

$$\therefore \quad \text{Safe } v = 1{\cdot}64\sqrt{\frac{1{\cdot}25 \times 2240}{{\cdot}26}} = 170 \text{ ft. per sec.}$$

This velocity is reckoned for radius R, which for a flywheel may be taken at the centre of the rim, but for a grindstone

$$R = \frac{\text{external radius}}{\sqrt{2}} = \cdot 7 \text{ of external radius.}$$

A much less velocity (about 80 feet per second) is adopted in practice.

Strength of Bolts.—In an ordinary bolt with V thread, the nut being deep enough, the bolt must break by a combination of tension and torsion, ·13 of the bolt area being devoted to resist the latter, according to Unwin. In practice both are allowed for by putting a small value on the safe stress—3 tons per sq. in. for Wrought Iron, and 4 tons for Steel, estimated on the area at thread bottom. Cylinder covers must be bolted very tightly, and an initial screwing stress often resisted also, so the working stress per square inch may be:

	Steel bolts.	W. I. bolts.
For 3 feet cylinders	4 tons	3 tons
For 2 feet cylinders	3 tons	$2\frac{1}{4}$ tons
For 1 foot cylinders	2 tons	$1\frac{1}{2}$ tons

The diameter at thread bottom may be found from p. 213 and p. 192. Thus a $\frac{3}{4}''$ bolt has a thread $\cdot 1''$ pitch, and depth of thread $= \cdot 1 \times \cdot 64 = \cdot 064$.

$$\therefore \text{ Dia. at thread bottom} = \cdot 75 - 2 \times \cdot 064 = \underline{\cdot 622}$$

$$\text{and area at thread bottom} = \frac{22 \times \cdot 311^2}{7} = \underline{\cdot 304}$$

No faced joints, except very small ones, should have bolts less than $\frac{3}{4}''$ dia., or they may be broken merely by screwing up,

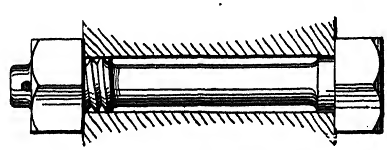


Fig. 354.

and their pitch should not be *greater* than six times the bolt diameter. In bolts that have to resist shock, the shank should be turned down, as in Fig. 354, to the diameter at thread bottom. (*See Appendix II., pp.* 833 *and* 842.)

[illegible]

Example [illegible]. A cylinder [illegible]″ dia. has a steam pressure of [illegible] lbs. per sq. in. Bolt circle [illegible]″ dia. and circumferential pitch to be [illegible] times bolt diameter. Find bolt diameter and number of bolts required.

[illegible]

Checking, we have

Strength of bolts = total load

[illegible] × number = [illegible]

and number of bolts = [illegible]

Finally pitch = [illegible] or [illegible] times bolt dia.

Compressive Stress-action, pure and simple, exists in few cases. The calculation is similar to that for tensional stress, and may be applied to all short columns whose length is only ten or twelve times the diameter. Thus

Example 10.—Find the thickness of a short, hollow, cast-iron column of 18″ outside diameter, to sustain a live load of 80 tons, plus a dead load of 100 tons. (Eng. Exam. 1888.)

$$\text{Equivalent dead load} = 100 + (2 \times 80) = 260 \text{ tons}$$

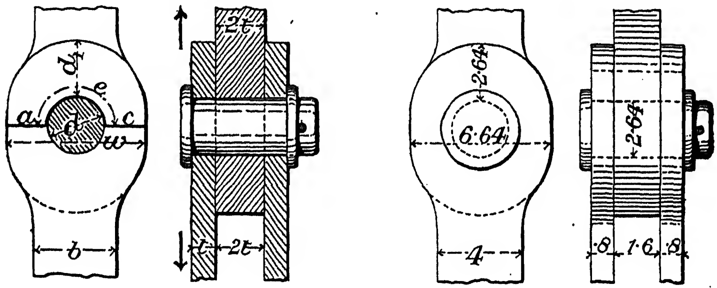
$$260 = 4\,a \qquad \text{and } \underline{a = 65 \text{ sq. ins.}}$$

$$\text{But } a = \pi R^2 - \pi r^2 = 65 \qquad \text{or } \frac{22}{7}(81 - r^2) = 65$$

$$\text{and } r = \sqrt{60 \cdot 3} = 7 \cdot 8 \qquad \therefore \ \underline{t = 9 - 7\cdot 8 = 1\cdot 2 \text{ ins.}}$$

Shear Stress-Action rarely occurs alone, but pins and rivets are thus calculated: $W = f_s\, a.$ *

Strength of a Suspension Link (see Fig. 355).—The strength of one thin link in tension, at a and c; the shear



Strength of a Suspension Link *Fig. 355.*

strength of the pin d; the strength at b; and the bearing stress on projected area of e, should each equal half the load:

$$\therefore \ \overset{(1)}{\frac{W}{2}} = \overset{(2)}{f_t\,(w - d)\,t} = \overset{(3)^*}{\tfrac{3}{4} f_s \frac{\pi d^2}{4}} = \overset{(4)}{f_t\, b\, t} = \overset{(5)}{f_b\, d\, t}$$

Let $f_t = 1$, $f_b = 1\frac{1}{2}$, and $f_s = \frac{3}{4}$.

By (4) and (5)	$1 \times b t = 1\frac{1}{2}\, d t$	and	$\underline{d = \cdot 66\, b}$
By (2) and (4)	$1\,(w - d)\, t = 1 \times b t$	and	$\underline{w = 1\cdot 66\, b}$
By (3) and (4)	$\frac{3}{4} \times \frac{3}{4} \frac{\pi d^2}{4} = 1\ b t$	and	$\underline{t = \cdot 20\, b}$
By actual tests	$d_1 = d$		$= \underline{\cdot 66\, b}$

and the thick link must be $2\,t$ in thickness.

(*See Appendix IV., p.* 952.)

* *See p.* 415.

Example 11.—A wrought-iron suspension bridge chain supports 32 tons. Find its dimensions and draw the joint to scale.

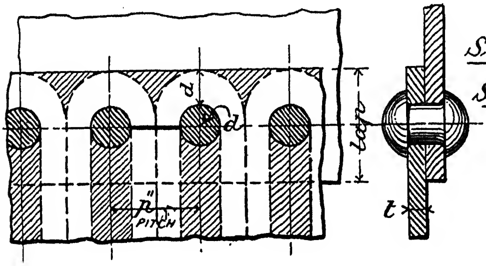
$$\text{Here } f_t\,(b \times \cdot 2\,b) = 16 \text{ tons} \quad \text{and } \underline{b = \sqrt{16} = 4}$$

$$\therefore \underline{d = \cdot 66 \times 4 = 2\cdot 64} \qquad \underline{d_1 = 2\cdot 64}$$

$$\underline{w = 1\cdot 66 \times 4 = 6\cdot 64} \qquad \underline{t = \cdot 2 \times 4 = \cdot 8}$$

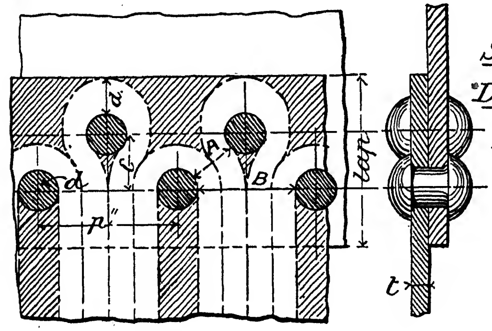
and the whole is drawn in Fig. 355.

Strength of Riveted Joints.—A boiler plate may be supposed to consist of similar links to the above, but with some redundant material between (see Figs. 356 and 357). The joint



Strength of Single-riveted Lap Joint

Fig. 356.



Strength of Double-riveted Lap Joint.

Fig. 357.

may give way by (1) shearing the rivet, (2) tearing the plate between rivets, (3) cross-breaking at d_1, and (4) crushing by reason of too thin a plate.

In a Single-riveted Lap Joint, as n Fig. 356, shear strength of one rivet = tensile strength of plate between two rivets,

$$\text{or} \quad f_s \frac{\pi d^2}{4} = f_t (p'' - d) t$$

which is our general formula. But the rivet (up to 1″ plates bears a definite proportion to the plate thickness, thus:

$$d = 1{\cdot}2 \sqrt{t} \text{ before riveting}$$

$$d_1 = 1{\cdot}3 \sqrt{t} \text{ after riveting,} \quad \text{and } t = {\cdot}6\, d_1^2$$

Also steel plates and rivets are the usual practice, where $f_s = 5$ and $f_t = 6$. Putting these in the formula, we have

$$\frac{5 \times 22 \times d_1^2}{7 \times 4} = 6\,(p'' - d_1)\,{\cdot}6\, d_1^2$$

$$\therefore \quad \underline{\text{pitch} = 1{\cdot}09 + d_1}$$

which shews that the space between rivets is a constant quantity for all plates up to 1″ thick. Also lap = 3 times d. (*See Appendices I. and III., pp.* 760 *and* 920.)

SIZES OF RIVETS AND PLATES (IN INCHES).

Plate thickness.	$\sqrt{t}$	Rivet.	Rivet hole.
$\frac{5}{16}$	·56	$\frac{11}{16}$	·73
$\frac{3}{8}$	·61	$\frac{3}{4}$	·8
$\frac{7}{16}$	·661	$\frac{13}{16}$	·86
$\frac{1}{2}$	·7	$\frac{7}{8}$	·91
$\frac{9}{16}$	·75	$\frac{15}{16}$	·975
$\frac{5}{8}$	·8	$\frac{15}{16}$	1·04
$\frac{11}{16}$	·83	1	1·08
$\frac{3}{4}$	·866	$1\frac{1}{16}$	1·125
$\frac{13}{16}$	·9	$1\frac{1}{16}$	1·17
$\frac{7}{8}$	·93	$1\frac{1}{8}$	1·2
$\frac{15}{16}$	·96	$1\frac{1}{8}$	1·25
1	1	$1\frac{1}{4}$	1·3
$1\frac{1}{8}$	1·06	$1\frac{1}{4}$	1·3
$1\frac{1}{4}$	1·1	$1\frac{1}{4}$	1·3

The Efficiency of joint has been already mentioned, and its value,

$$\eta = \frac{\text{strength of pierced plate}}{\text{strength of solid plate}} = \frac{p'' - d_1}{p''}$$

$\therefore$ for single riveted joint,

$$\text{with } \tfrac{3}{8}'' \text{ plate, } \eta = \frac{1{\cdot}09}{1{\cdot}09 + {\cdot}8} = \underline{{\cdot}57 \text{ or } 57\%}$$

$$\text{and with } 1'' \text{ plate, } \eta = \frac{1{\cdot}09}{1{\cdot}09 + 1{\cdot}3} = \underline{{\cdot}45 \text{ or } 45\%}$$

The Strength of a Double-riveted Lap Joint (zigzag) can easily be discussed by reference to the 'virtual links' in Fig. 357. Clearly plate A must equal one rivet, while B equals two rivets, in strength. So the centres at A will be $1{\cdot}09 + d_1$, while those at B (called the pitch),

$$\underline{p'' = 2\,(1{\cdot}09) + d_1 = 2{\cdot}18 + d_1}$$

$$\text{and} \quad C = \sqrt{(A + d_1)^2 - \left(\frac{B + d_1}{2}\right)^2}$$

The distance from rivet centre to plate edge will be $1\tfrac{1}{2}\,d$ as before, deduced from practice

$$\text{and the efficiency } \eta = \frac{2{\cdot}18}{2{\cdot}18 + d_1}$$

$$\text{For } \tfrac{3}{8}'' \text{ plate } \eta = \frac{2{\cdot}18}{2{\cdot}18 + {\cdot}8} = \underline{{\cdot}73 \text{ or } 73\%}$$

$$\text{For } 1'' \text{ plate } \eta = \frac{2{\cdot}18}{2{\cdot}18 + 1{\cdot}3} = \underline{{\cdot}625 \text{ or } 62\%}$$

Example 12.—Find the various dimensions of lap joints for $\tfrac{3}{8}''$ boiler plates; (1) single riveted, and (2) double riveted.

(1) $d = \tfrac{3}{4}''$ and $\underline{d_1 = {\cdot}8 \text{ or } \tfrac{13}{16}''}$

$\underline{p'' = 1{\cdot}09 + d_1 = 1\tfrac{7}{8}''}$

$\underline{\text{lap} = 3 \times {\cdot}8 = 2\tfrac{7}{16}''}$

and the joint is drawn at Fig. 358.

[illegible]

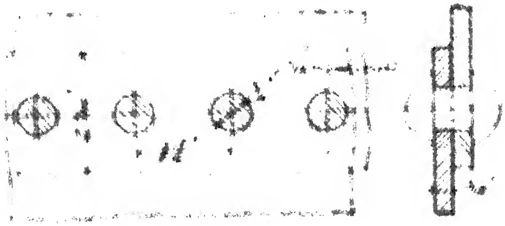


Fig. 358

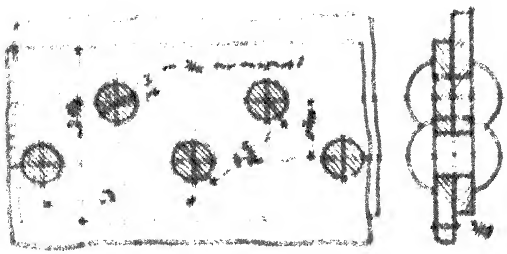


Fig. 359

Example 13. Find the pitch of a single-riveted lap joint where $d = \frac{3}{4}''$, $t = \frac{1}{2}''$, $f_s = 21000$ lbs. and $f_t = 19,000$ lbs. (Eng. Exam. 1886.)

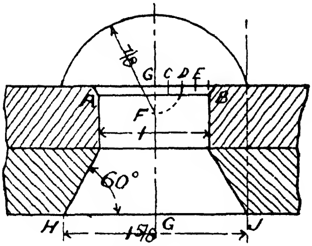
$$f_s \frac{\pi d^2}{4} = f_t (p - d) t$$

$$\frac{21000 \times 22 \times 3 \times 3}{7 \times 4 \times 4 \times 4} = 19,000 \left(p - \tfrac{3}{4}\right) \tfrac{1}{2} \qquad \underline{p = 1.81 \text{ or } 1\tfrac{13}{16}''}$$

Contour of Rivet Head. Fig. 360 shows how to draw a cup or snap head, and a countersunk. Mark out plate thickness and rivet diameter. Divide A B, by guesswork, into four equal parts, and at centre C strike arc D E. Lastly, with centre F and radius A E strike the cup curve. Both snap head and countersunk have a diameter of $1\frac{3}{4}\,d$.

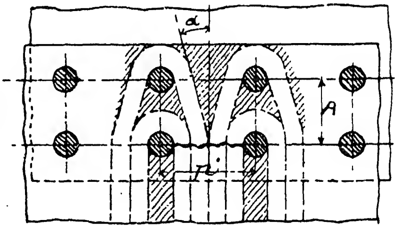
Strength of Chain-riveted Lap Joint. The same elements are required as for zigzag riveting, for the same links have to pass. Centres A, Fig. 361, may be $1\frac{3}{4} \times d$, so that the angle a

may not be too large. The efficiency will be reckoned on the pitch line as before, because the joint is weakest along that line,



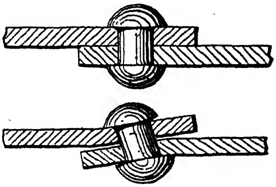
Rivet Head.

Fig. 360.

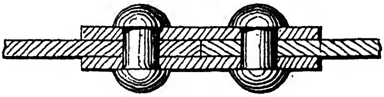


Chain-riveted Lap Joint

Fig. 361.

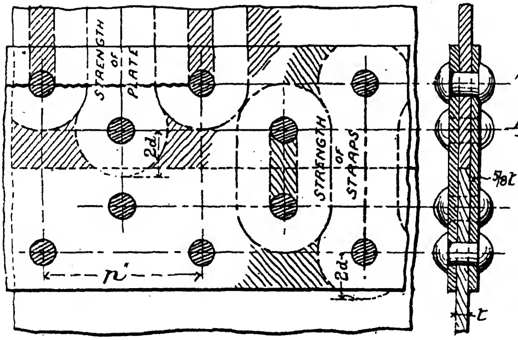


LAP JOINT



BUTT JOINT

Fig. 362.



Strength of Double-riv^d Butt Joint.

Fig. 363.

the *links being most crowded* there. The fracture should always be allowed to take place preferably on the pitch line.

All the seams given may have two butt straps instead of a lap, and the dangerous bending action be thereby removed, see Fig. 362.

As each rivet is then in double shear, and twice the previous strength, the pitch may be considerably increased, but this cannot be taken advantage of, except perhaps in the thickest plates, or staunchness would be affected.

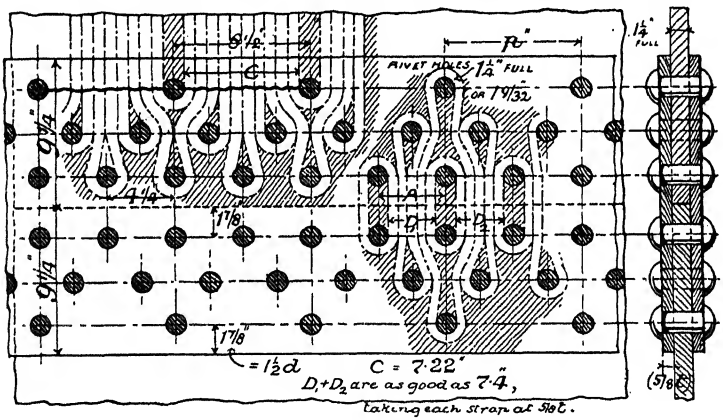
A Double-riveted Butt Joint with two Coverplates is shewn in Fig. 363, designed to use the full strength of rivets. Of course the links will be twice the width of a lap joint,

$$\therefore p'' = 4\ (1\cdot 09) + d_1 = 4\cdot 36 + d_1$$

$$\text{and} \quad \text{diagonal centres} = 2\ (1\cdot 09) + d_1 = 2\cdot 18 + d_1.$$

The butt strap might be $\frac{1}{2}t$ in thickness, but is safer at $\frac{5}{8}t$. It might have to be thicker and should always be examined separately, as one plate equal to the two straps put together. The overlap may also have to be increased as at $2d$.

The Treble-riveted Butt Joint at Fig. 364 is taken from



Strength of Treble-riveted Butt Joint Fig. 364.

marine practice. Using the *general formula* (page 407), $(p - d) = w''$ $= \cdot 838''$; but as 5 links must pass at the pitch line:

Examining as a lap joint, width of one link $= \cdot 838''$

$$\text{and} \quad p'' = 5\ (\cdot 838) + 1\cdot 28 = 5\cdot 47''$$

Examining as a butt joint, width of one link $= \cdot 838 \times 2 = 1\cdot 676''$

$$\text{and} \quad p'' = 5\ (1\cdot 676) + 1\cdot 28 = 9\cdot 66''$$

the intermediate value $8\frac{1}{2}''$ having been taken. Next, the butt straps must pass $2\frac{1}{2}$ rivets each at D_1 and D_2, or $D_1 + D_2$ must pass the same strength as C. But

$$\text{Plate at } C = 7{\cdot}22t.$$

$$\text{and Plate at } D_1 + D_2 = 5{\cdot}94 \times 1{\cdot}25t = 7{\cdot}4t,$$

or the links are most crowded at the pitch line.

Taking now the joint as designed,

Strength of pierced plate $= (8{\cdot}5 - 1{\cdot}28)\ 1{\cdot}28 \times 6 = 55$ tons.

Strength of rivets $= \dfrac{5 \times 22 \times 1{\cdot}28 \times 1{\cdot}28}{7 \times 4} \times 5 \times 2 = 64{\cdot}34$ tons.

Strength of solid plate $= 8{\cdot}5 \times 1{\cdot}28 \times 6 = 65{\cdot}28$ tons.

$$\text{and} \quad \eta = \frac{55}{65{\cdot}28} = {\cdot}8425 \text{ or } 84\tfrac{1}{4}\ \%$$

The Tie Bar or Stringer Plate, Fig. 365, is an important deduction from the last example. By compelling the joint to break

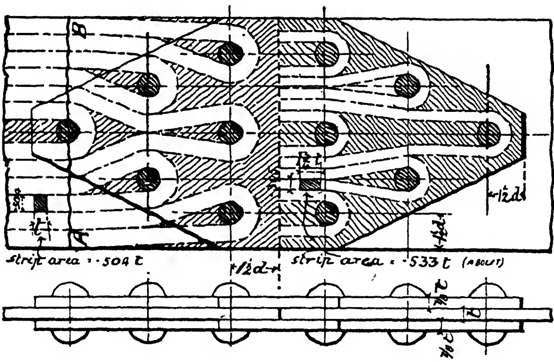


Fig. 365.

preferably at AB, *the plate is only weakened to the extent of one rivet.* The strips must not be bent abruptly, however, and the butt straps should always be examined separately, and their thickness increased until the links are narrowed sufficiently for all to pass; thus $\frac{7}{8}t$ is required in the example.

VALUES OF η FOR VARIOUS THICKNESSES OF PLATE AND VARIOUS FORMS OF JOINT, USING STEEL PLATES AND STEEL RIVETS, AND FULL STRENGTH OF BUTT JOINTS.

t	Dia. of rivet hole.	Single-riveted lap.		Single-riveted butt, with 2 straps.		Double-riveted lap.		Double-riveted butt, with 2 straps.		Treble-riveted with 1 strap.		Treble-riveted with 2 straps.	
		w'' = 1 rivet.	η	w'' = 2 rivets.	η	w'' = 2 rivets.	η	w'' = 4 rivets.	η	w'' = 5 riv.	η	w'' = 10 rivets.	η
$\frac{5}{16}$	·73	1·09	·59	2·18	·75	2·18	·75	4·36	·85	5·45	·88	10·9	·93
$\frac{3}{8}$	·8	1·09	·57	2·18	·73	2·18	·73	4·36	·84	5·45	·87	10·9	·93
$\frac{7}{16}$	·86	1·09	·56	2·18	·71	2·18	·71	4·36	·83	5·45	·86	10·9	·92
$\frac{1}{2}$	·91	1·09	·54	2·18	·70	2·18	·70	4·36	·82	5·45	·85	10·9	·92
$\frac{9}{16}$	·975	1·09	·53	2·18	·69	2·18	·69	4·36	·81	5·45	·84	10·9	·91
$\frac{5}{8}$	1·04	1·09	·51	2·18	·68	2·18	·68	4·36	·80	5·45	·83	10·9	·91
$\frac{11}{16}$	1·08	1·09	·50	2·18	·67	2·18	·67	4·36	·79	5·45	·83	10·9	·91
$\frac{3}{4}$	1·125	1·09	·49	2·18	·66	2·18	·66	4·36	·79	5·45	·82	10·9	·90
$\frac{13}{16}$	1·17	1·09	·48	2·18	·65	2·18	·65	4·36	·78	5·45	·82	10·9	·90
$\frac{7}{8}$	1·2	1·09	·47	2·18	·64	2·18	·64	4·36	·78	5·45	·82	10·9	·90
$\frac{15}{16}$	1·25	1·09	·46	2·18	·63	2·18	·63	4·36	·78	5·45	·81	10 9	·90
1	1·3	1·09	·45	2·18	·62	2·18	·62	4·36	·77	5·45	·80	10·9	·90
$1\frac{1}{16}$	1·3	1·034	·44	2·068	·61	2·068	·61	4·136	·76	5·17	·79	10·3	·90
$1\frac{1}{8}$	1·3	·976	·43	1·958	·60	1·952	·60	3·914	·75	4·88	·79	9·96	·88
$1\frac{3}{16}$	1·3	·925	·41	1·850	·57	1·850	·57	3·700	·74	4·62	·78	9·25	·87
$1\frac{1}{4}$	1·3	·878	·40	1·756	·57	1·756	·57	3·512	·73	4·39	·77	8·78	·87

Remarks.—In cooling, the rivet exerts great grip on the plate, giving frictional strength to the joint, but caulking diminishes this, so it is not allowed for. Rivets over 6″ long would break in cooling, so must be hammered up cold.

The formula for boiler strength, $p^{\text{tons}}\, r = ft\, \eta$, can now be used to better advantage. Construct a table shewing t and η under all conditions, and after finding $t \times \eta$ from the formula, choose such values of each as will meet the case when multiplied. Such a table precedes this page, where the pitch has been taken at its theoretical value; but must be decreased to secure staunchness where necessary, as with the thinner plates in the last column. *The efficiency table is not from practical tests, but from p.* 407 *formulæ.*

Example 14.—A steel Lancashire boiler 8 ft. dia. is loaded with 100 lbs. per square inch. Find t, and indicate the joint you would use.

$$p^{\text{tons}}\, r = f_t\, t\, \eta \quad \therefore\ t\,\eta = \frac{100 \times 4 \times 12}{2240 \times 6} = \cdot 357$$

1. Single riveted lap joint $\quad t\eta = \dfrac{3}{4} \times \cdot 49 = \cdot 367$

2. Single riveted butt joint
3. Double riveted lap joint $\quad \Big\}\ t\eta = \dfrac{9}{16} \times \cdot 69 = \cdot 382$

4. Double riveted butt joint $\quad t\eta = \dfrac{7}{16} \times \cdot 83 = \cdot 363$

Something between (3) and (4) would have to be adopted; say (4) with $\frac{1}{2}$″ plate and spacing like (3) for staunchness.

Example 15.—Two lengths of mild steel tie rod 7″×1″ are to be connected with double butt straps. Determine dimensions and efficiency. (Hons. Mach. Constr. Exam., 1893.)

$$d_1 = 1\cdot 3. \quad \text{Centre to edge} = 1\cdot 5 \times 1\cdot 25 = 1\cdot 875$$

$$w'' \text{ for one rivet, in double shear} = 2 \times 1\cdot 09 = 2\cdot 18$$

$$7 - 1\cdot 3 = 5\cdot 7 \text{ width of pierced plate}: \quad \therefore\ \frac{5\cdot 7}{2\cdot 18} = \underline{3 \text{ rivets say.}}$$

Checking we have:

$$\text{Strength of rivets} = 2 \times \frac{\pi d_1^2}{4} f_t \times 3 = \frac{2 \times 22 \times 1\cdot 3 \times 1\cdot 3 \times 5 \times 3}{7 \times 4} = \underline{39\cdot 83 \text{ tons}}$$

$$\text{Strength of pierced plate} = (7 - 1\cdot 3) \times 1 \times 6 = \underline{34\cdot 2 \text{ tons}}$$

[illegible]

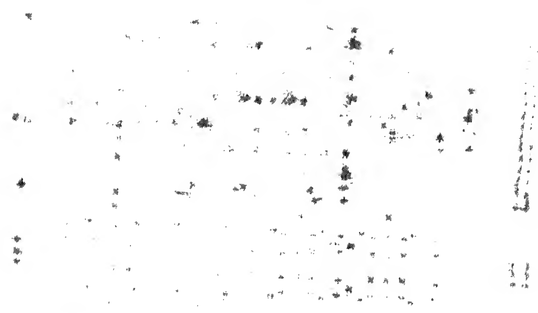


Fig. 165.

Strength of Pins and Bolts in Shear. Unless a pin [illegible] the [illegible], as in good [illegible] [illegible], and [illegible] safe stress must be [illegible]:

[illegible] round sections ● × safe stress $-\frac{?}{?}f_s$

[illegible] square sections ■ × safe stress $-\frac{?}{?}f_s$

[illegible] diagonal sections ◆ × safe stress $-\frac{?}{?}f_s$

Strength of Cotter Joint. Fig. 165 shews this joint connecting two lengths of [illegible] rod. It may break (1) by shearing the cotter, (2) and (3) by crushing the cotter, (4) by tearing the socket, (5) by tearing the [illegible] rod, and (6) by tearing the [illegible] rod. Supposing all parts be made of equal strength and of forged steel, where f_t = [illegible] tons, f_s = [illegible] tons, and f_c = 14 tons,* we have

(1) Strength of cotter for shear = [illegible]
(2) " cotter for bearing on [illegible] rod = 14 $d_1 t$
(3) " cotter for bearing on socket = 14 [illegible] $d_1 t$
(4) " socket = [illegible]

* f_c may be much greater than f_t if the material be well [illegible] all round.

(5) Strength of solid rod $= 7 \times \frac{22}{7 \times 4} d^2 \quad = \frac{11}{2} d^2$

(6) " pierced rod $= 7\left(\frac{22}{7 \times 4} d_1^2 - d_1 t\right) = \frac{11}{2} d_1^2 - 7 d_1$

(7) Strength at b_2 for shear $= 5 \times 2 \times b_2 d_1 \quad = 10 b_2 d_1$

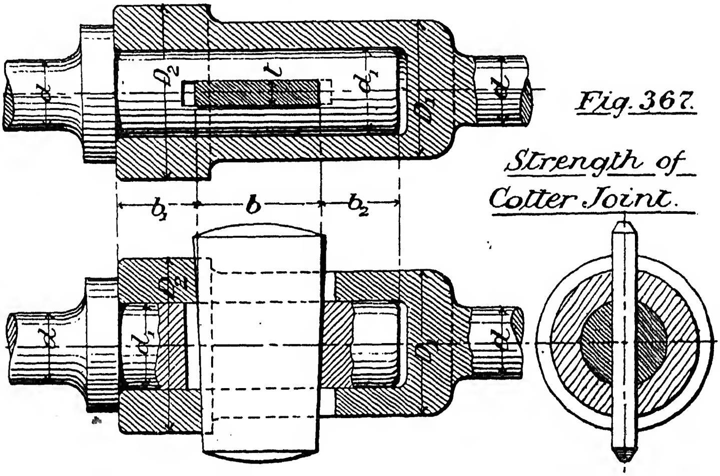


Fig. 367.

Strength of Cotter Joint.

Equating, we obtain:

By (2) and (5): $14 d_1 t = \frac{11}{2} d^2$ and $d_1 t = \cdot 393 d^2$ ...... (8)

By (6), (5), and (8) $\Big\}$ $\frac{11}{2} d_1^2 - 7 d_1 t = \frac{11}{2} d^2 \quad \therefore d_1 = \sqrt{1 \cdot 5 d^2} = 1 \cdot 22 d$ (9)

By (2) and (5): $14 d_1 t = \frac{11}{2} d^2 \quad \therefore t = \cdot 322 d$ .... (10)

By (1), (5), and (10): $10 b t = \frac{11}{2} d^2 \quad \therefore b = 1 \cdot 71 d$ .... (11)

By (4), (5), (9), and (10) $\Big\}$ $\frac{11}{2}(D_1^2 - d_1^2) - 7t(D_1 - d_1) = \frac{11}{2} d^2 \therefore D_1 = 1 \cdot 62 d$ (12)

By (3) and (5), (9) and (10) $\Big\}$ $14(D_2 - d_1)t = \frac{11}{2} d^2 \therefore D_2 = 2 \cdot 44 d$ or $2 d_1$ (13)

By (7) and (5): $10 b_2 d_1 = \frac{11}{2} d^2 \quad \therefore b_2 = \cdot 45 d$ .... (14)

The values at (12) and (14) are both unreasonably small, and are increased in practice to $D_1 = 2 d$; and $b_1 = b_2 = \frac{3}{4} b = 1 \cdot 28 d.$

Example 16.—A foundation bolt with square head (Fig. 368) is secured by a cotter. Find D, b, and t in terms of d, where f_t, f_s, and f_b vary as 1 : $\frac{3}{4}$: 2 respectively. (Hons. Mach. Constr. Exam. 1886.)

Following the previous calculations:

$$\underline{D = 1{\cdot}08\,d,} \quad \underline{b = 1{\cdot}44\,d,} \quad \text{and } \underline{t = {\cdot}363\,d.}$$

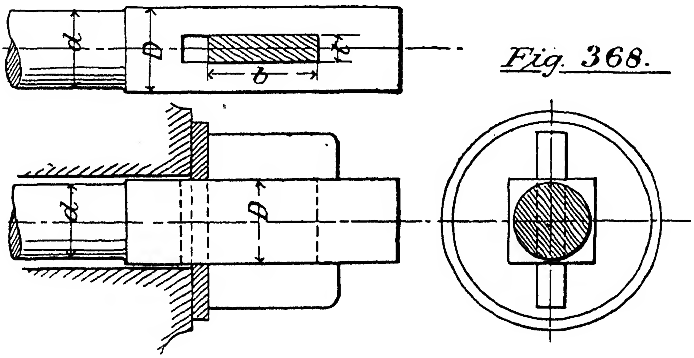


Fig. 368.

Torsional Stress-Action unallied with bending occurs only in very short shafts. In any case the two actions must be separately considered. Fig. 369 shews a shaft under twist, the external load being caused by the couple $d \times b\,c$, while the internal resistance of the shaft is shewn by the couple $e \times f\,g$.*

$\therefore$ External moment = moment of resistance of section

$$\text{or} \quad Wr = f_s Z_t$$

where Z_t, the modulus of section, is a number depending on the size and shape of the section.

Strength of Solid Round Shaft.—Let r be the outer radius of a solid shaft, Fig. 370. Imagining the section divided into concentric rings:

$$\text{Total stress on outer ring} = f_s \times 2\pi r \times t \qquad (1)$$

But f_s diminishes towards the centre because δ_s decreases:

$$\therefore \text{ Total stress at any other ring } r_1 = \frac{r_1}{r} f_s \times 2\pi r_1 \times t \qquad (2)$$

* A couple is formed by a pair of equal and opposite forces, and can produce turning effect only, being represented by 'one force × total arm.'

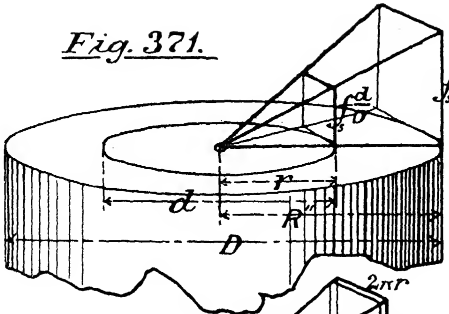


Fig. 371.

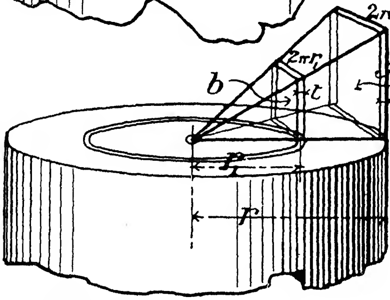


Fig. 370.

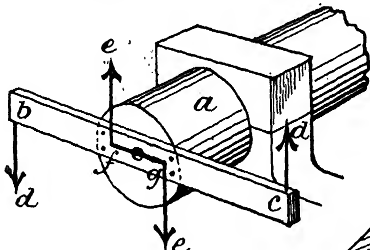
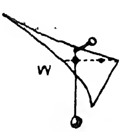


Fig. 369.



Strength of Shafts.

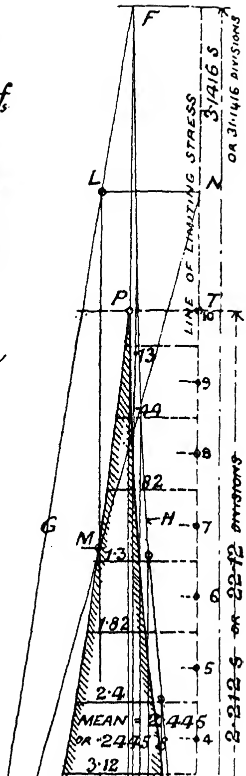
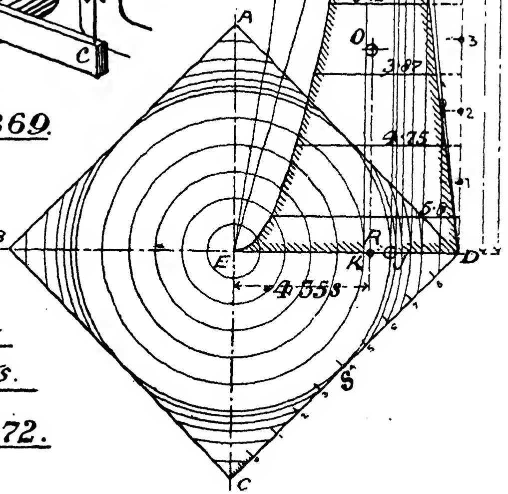



Fig. 372.

The first formula may be represented by the lamina at a, and the second by that at b, and the total of the stresses on *all* the rings will be given by the pyramid. Again:

$$\text{Moment of stress at ring } r = a \times r$$
$$\text{Moment of stress at ring } r_1 = b \times r_1$$

$$\therefore \text{Moment of all the stresses} = \text{contents of pyramid} \times \text{average arm}$$
$$= (\text{base} \times \tfrac{1}{3} \text{ height}) \times (\tfrac{3}{4} \text{ height})$$

$$\therefore \underline{\text{Moment of resistance of section}} = f_s \frac{2\pi r \times r}{3} \times \frac{3}{4} r = f_s \frac{\pi r^3}{2}$$

$$\text{(and putting } \frac{d}{2} = r) \quad = \underline{f_s \left(\frac{\pi d^3}{16} \right)}$$

Strength of Hollow Round Shaft.—Fig. 371 shews a shaft of diameters D and d, externally and internally respectively. At radius R″ the stress is f_s, but at radius r it is proportionately less, being but $f_s \frac{d}{D}$. The strength of the hollow shaft will be found by deducting the strength of shaft d, as stressed *in situ*, from the strength of a solid shaft D.

$$\therefore \underline{\text{Moment of resistance}} = \text{mom}^t \text{ of solid shaft } \textit{minus} \text{ mom}^t \text{ of core}$$

$$= f_s \frac{\pi D^3}{16} - \left(f_s \frac{d}{D} \right) \frac{\pi d^3}{16} \qquad = \underline{f \frac{\pi}{16} \left(\frac{D^4 - d^4}{D} \right)}$$

(*See App. III., p.* 921.)

Strength of Square Shaft.—In this case we shall not use the previous methods, but shall adopt a construction which, although requiring careful drawing, can be employed for any section, and is therefore general. In Fig. 372, ABCD is the shaft section, divided into concentric rings as before. Erect perpendiculars on ED to represent the length of every ring, and bound these by the figure EGLFD. JF $= \pi$ S, and FE is a straight line, while the lengths between F and D are found by stepping off each set of four arcs with dividers. Now the stress will be greatest at D, and will decrease gradually to zero at E, and the product ($f_s \times$ ring area) will proportionately decrease, so the total stress may be obtained by imagining f_s to be constant, and each ring to have a value represented by the circumference decreased

according to its distance from D, the point of greatest stress. Thus, if the ring K L be projected to D N, and N E joined, the length K M will represent the virtual length of the ring if f_s be constant. Treating every perpendicular similarly, we obtain the curve E P D, and the shaded figure is the *virtual stress area*, or area of equal stress. Now cut out a copy of the shaded figure in thin cardboard, and, hanging loosely from a pin in two different positions, as at W, mark plumb lines from the pin in each case upon the paper. The crossing point will be O, the centre of gravity, or centre of all the stresses, and the arm = E R. Next, find the area of the figure. Divide s into 10 parts, and *measure everything in terms of these parts.* Divide D T into 10 parts, and draw horizontals from the middle of each part; then measure their intercepts on the figure. Adding all these figures (·13, ·44, ·82, &c.) and dividing by 10 we get the mean width 2·445, or ·2445 s. The height D T measures 22·12 parts, or 2·212 s, so the area = height × mean width, and

$$\underline{\text{Moment of resistance of section}} = f_s \times \text{stress area} \times \text{arm}$$

$$= f_s \times \text{height} \times \text{mean width} \times \text{arm}$$

$$= f_s \times 2{\cdot}212\, s \times {\cdot}2445\, s \times {\cdot}435\, s \quad = \underline{f_s({\cdot}235\, s^3)}^*$$

St. Venant shewed, however, in 1856, that the previous methods (Coulomb's ring theory) were not strictly applicable to any but circular sections, and gave the following:

$$\underline{\text{Moment of square section} = f_s({\cdot}208\, s^3)} \text{ or } {\cdot}88 \text{ of } \left\{f_s({\cdot}235\, s^3)\right\}$$

because the greatest stress occurs at the middle of each side. To illustrate St. Venant, Thomson and Tait have imagined the shaft to be a box full of liquid, which, if rotated, would leave the latter behind somewhat, and the apices would cause two stresses—tangential and centripetal—to act on the particles, the former only being of momental value.

* Generally $T_m = f_s \dfrac{I}{y}$, $Z_t = \dfrac{I}{y}$, and $I = Z_t\, y$, where I is the polar moment of inertia (see p. 429).

The Strength of a Rectangular Section was given by St. Venant as follows:

Modulus of resistance of rectangular section $= \cdot 2944 \dfrac{b^2 h^2}{\sqrt{b^2+h^2}}$

while the pure ring theory would give $\cdot 1666\ b\ h \sqrt{b^2+h^2}$, and the discrepancy increases with the ratio $\frac{h}{b}$

Thus if $b = 1''$ and $h = 2''$

$T_m = \cdot 5266$ ton ins. (1) by St. Venant; and $\cdot 745$ ton ins. [(2) by ring theory,

and (1) $= \cdot 7$ of the diagram value (2).

If $b = 1''$ and $h = 4''$

$T_m = 1 \cdot 142$ (1) and $2 \cdot 747$ (2) respectively

or (1) $= \cdot 41$ of diagram value.

Hexagonal shafts may be treated directly from diagram.

Strength of Shafts by Direct Experiment.—The following experimental figures may be used by way of correction. Moment of any shaft d_1 or $s =$ Figure in table $\times$ (d^3 or s^3).

BREAKING MOMENTS OF SHAFTS 1" DIA. AND 1" SQUARE, IN TON-INCHES (ELSWICK EXPERIMENTS).

	Round.	Square.
Wrought Iron	5·35	6·83
Cast Iron...	5·31	6·78
Steel...	8·92	11·6
Yellow Brass	2·45	3·15
Cast Copper	2·15	2·74

A factor of 10 is to be used for short shafts and of 16 for long shafts, to secure stiffness. Strength is rarely the sole criterion.

Example 17.—Find the relative weights of two shafts of equal strength; the one solid, and the other hollow, with a hole half the outside diameter. (Eng. Exam. 1892.)

Moment of solid shaft = moment of hollow shaft.

$$f_s \frac{\pi}{16} d_1^3 = f_s \frac{\pi}{16}\left(\frac{D^4 - d^4}{D}\right) \qquad \therefore\ d_1 = \sqrt[3]{\frac{D^4-(\frac{1}{2}D)^4}{D}}$$

Let D = 1. Then $d_1 = \sqrt[3]{\frac{1-\frac{1}{16}}{1}} = \sqrt[3]{\frac{15}{16}} = \cdot 979$

$$\frac{\text{Weight of solid shaft}}{\text{Weight of hollow shaft}} = \frac{\pi r_1^2}{\pi(R^2 - r^2)} = \frac{\cdot 979^2}{1^2 - (\frac{1}{2})^2} = \underline{1\cdot 277 : 1}$$

Example 18.—Find the relative strengths of shafts :—
$2\frac{1}{2}''$ round, $3\frac{1}{2}''$ round, $3''$ square, and $5'' \times 2''$ rectangular.

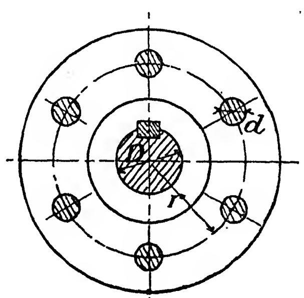
Moment of $2\frac{1}{2}''$ round $\propto \frac{\pi}{16} d^3 = \cdot 1963 \times 15\cdot 62 = \underline{3\cdot 066 \text{ say } 31.}$

" $3\frac{1}{2}''$ " $\propto$ " $= \cdot 1963 \times 42\cdot 87 = \underline{8.415 \text{ say } 84.}$

" $3''$ square $\propto \cdot 208\, s^3 = \cdot 208 \times 27 = \underline{5\cdot 616 \text{ say } 56.}$

" $5'' \times 2''$ rect. $\propto \cdot 2944 \frac{b^2 h^2}{\sqrt{b^2+h^2}} = \underline{5\cdot 467 \text{ say } 55.}$

Strength of Coupling Bolts.—Fig. 373 is the face view of



Strength of Coupling Bolts.
Fig. 373.
X = no. of bolts.

a flange coupling. As the bolts and shaft must be equally strong: and the allowable stress on the bolts $= \frac{3}{4} f_s$ (*see p.* 415)

Moment of bolts = moment of shaft

$$\left(\frac{3}{4} f_s \frac{\pi d^2}{4} X\right) r = f_s \frac{\pi}{16} D^3$$

$$\underline{d = \sqrt{\frac{D^3}{3\,X r}} = \cdot 577 \sqrt{\frac{D^3}{X r}}} \quad (\textit{See App. II. p. } 844.)$$

[illegible] flange [illegible] Mark [illegible]

[illegible]

[illegible] shafts, then mean [illegible] A the distortion of [illegible] [illegible] and [illegible] the [illegible] other B [illegible] twice [illegible]

Stress = I_a

In shaft A [illegible] [illegible]

[illegible]

[illegible] $= .06$ [illegible]

[illegible] shaft B, [illegible]

[illegible] and $f_s = .14\,f$

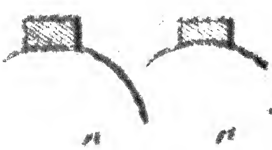
[illegible] greatest I_a

[illegible] assumed mean $I_a = 1$

Then, for A, [illegible] or as [illegible] exactly

and, for B, [illegible]

Strength of Sunk Keys may be investigated, though *flat keys* (a, Fig. 374) and *saddle keys* (a, Fig. 374) cannot be satis-



a a

Fig. 374

Strength of Keys

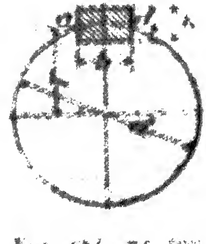


Fig. 375

factorily determined. Referring to Fig. 375, we must make (1) bearing surface at k, (2) shear through $a\,d$, and (3) shaft d, all of equal strength. Taking $f_s = f_c = \frac{1}{2}$.

Moment of shaft = moment of key (bearing) = moment of key (shearing)

$$f_s \frac{\pi d^3}{16} = f_b \frac{h}{2}\frac{l\,d}{2} = f_s\, b\, l \frac{d}{2}$$

(3) (1) (2)

By (3) and (1) $\frac{3}{4} \times \cdot 1963\, d^3 = \frac{2}{4} h l d \quad \therefore\ hl = \cdot 2944\, d^2$ (4)

By (1) and (2) $\frac{2\, hl\, d}{4} = \frac{3}{4}\, bl\, \frac{d}{2} \quad \therefore\ \underline{h = \frac{3}{4} b}$ .... (5)

Let $\underline{b = \cdot 3\, d}$ Then by (5) $\underline{h = \cdot 75 \times \cdot 3 = \cdot 225\, d}$

By (4) $hl = \cdot 2944\, d^2$ and $\underline{l = 1 \cdot 3\, d}$

In practice the following rules are adopted :

$$\underline{b = \tfrac{1}{4}\, d + \tfrac{1}{8}''} \text{ and } \underline{h = \tfrac{1}{8}\, d + \tfrac{1}{8}''}$$

$$\text{Then } \underline{l = \frac{\cdot 2944\, d^2}{\frac{1}{8} d + \frac{1}{8}}}$$

Angle of Torsion, or the angle through which one end of a shaft turns relatively to the other under a given stress.

$$C = \frac{\text{shear stress}}{\text{shear strain}} = \frac{f_s^{\text{lbs}}}{\delta_s} \quad \text{and } \delta_s = \frac{f_s^{\text{lbs}}}{C}$$

Referring to Fig. 376 and putting θ in circular measure $\left(\text{viz. } \frac{\text{arc}}{\text{rad.}}\right)$

$\theta = \frac{\delta_s}{r}$ for every inch of shaft length. Substituting $\frac{f_s^{\text{lbs}}}{C}$ for δ_s

$$\underline{\theta \text{ total} = \frac{f_s^{\text{lbs}}}{C\, r} \times l = \frac{2 f_s^{\text{lbs}}\, l}{C\, d}}$$

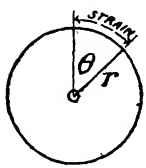


Fig. 376.

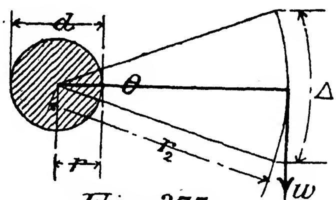


Fig. 377.

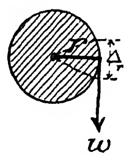


Fig. 378.

If a weight w produce a twist θ (Fig. 377), then

$$\theta = \frac{\Delta}{r_2} = \frac{2 f_s^{\text{lbs}}\, l}{C\, d} \qquad \text{But } w\, r_2 = f_s^{\text{lbs}} \frac{\pi d^3}{16} \text{ and } f_s^{\text{lbs}} = \frac{16\, w\, r_2}{\pi d^3}$$

$$\Delta = \frac{2 f_s^{\text{lbs}}\, l\, r_2}{\mathrm{C}\, d} = \frac{2\, w\, r_2 \times 16\, l\, r_2}{\pi d^3\, \mathrm{C}\, d} = \underline{\frac{w\, r_2^{\,2}\, l}{\mathrm{C}\, d^4} \times 10{\cdot}18}$$

...an be referred to radius r (Fig. 378) if w be increased.

...at strength $\mathrm{T_m} \propto d^3$, while stiffness $\frac{\mathrm{I}}{\theta} \propto d^4$.

...ple 21.—The angle of torsion of a round W. I. shaft is to ...ree for every 3 feet of length, and the maximum stress is ... lbs. per sq. in. Find the one diameter to satisfy both con-...ions. Mach. Constr. Exam. 1889.) (*See App. V., p.* 997.)

$$\theta = \frac{2 f_s^{\text{lbs}}\, l}{\mathrm{C}\; d} \quad \text{or} \quad \frac{22 \times 2}{7 \times 360} = \frac{2 \times 8000 \times 36}{10{,}500{,}000, \times d} \quad \therefore\ \underline{d_1 = 3{\cdot}14''}$$

...ple 21*a*.—The angle of torsion being limited to one degree ...10 feet of length, find the diameter of shaft to transmit ...at 50 revs. per m. (Hons. Applied Mech. Exam. 1892.)

...s above $\quad \frac{22 \times 2}{7 \times 360} = \frac{2 \times f \times 120}{10{,}500{,}000 \times d} \quad$ and $f = 763{\cdot}8\, d$

...y *p*. 508 $\quad d^3 = 320{,}810 \frac{100}{f \times 50} \quad$ and $f = \frac{641{,}620}{d^3}$

$$\therefore \quad 763{\cdot}8\, d = \frac{641{,}620}{d^3} \qquad \text{and} \qquad \underline{d = 5{\cdot}38}$$

...ngth of Helical (Spiral) Springs.—In the round ...ing (Fig. 379) the pull is exerted axially, and any ...*l* is in torsion.

$$\therefore \quad \underline{w\, r = f_s^{\text{lbs}} \frac{\pi d^3}{16}}$$

Extreme elastic stress for steel $= 89{,}000$ lbs.

and working stress $= \frac{89{,}000}{3} = 29{,}600$ lbs.

...quare-sectioned steel (Fig. 380),

$$\underline{w\, r = f_s^{\text{lbs}}\, ({\cdot}208\, d^3)}\,.$$

...ectangular sections,

$$\underline{w\, r = f_s^{\text{lbs}} \left({\cdot}2944 \times \frac{b^2\, h^2}{\sqrt{b^2 + h^2}} \right)} \quad (\textit{See p. } 421.)$$

Deflection of Helical Springs.—This may be found by imagining the wire uncoiled, and treated as a straight shaft. Let l = length of wire from A to B, and n = number of coils in that length (Fig. 379).

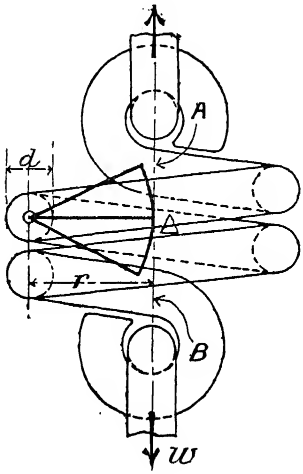


Fig. 379.

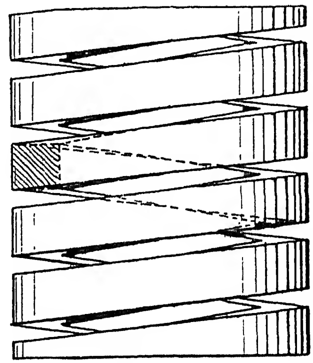


Fig. 380.

Helical Springs.

$$l = 2\pi r n, \quad \text{and} \quad \Delta = \frac{2 f_s^{\text{lbs}}\, l\, r}{C\, d} = \frac{2 w r}{\pi d^3} \frac{16 \times 2 \pi r n r}{C\, d} = \frac{w n r^3}{C\, d^4} 64$$

N.B.—This is for round wire only. For square wire,

$$\Delta = \frac{w n r^3}{C\, d^4} 60{\cdot}5$$

and for rectangular wire,

$$\Delta = \frac{w n r^3}{C\left(\dfrac{b^3 h^2}{\sqrt{b^2 + h^2}}\right)} 42{\cdot}6$$

(*See App. II., p.* 845.)

The above formulæ have been thoroughly tested for steel with C = 12,000,000 lbs. and found reliable with that value. The curve of work during extension or compression is found as for a bar (page 367).

Bending Stress-Action.—Fig. 381 represents a model devised by Prof. Perry to shew the stresses occurring in a beam. Supposing W very heavy and beam l so light as to be negligible,

W causes a bending moment or turning effect round A equal to $W \times l$, and also exerts a downward pull to be balanced by weight W_1, so that $W = W_1$. The latter is called the shearing force, and is felt on every vertical section of the beam. W_1 and W really form a couple with the arm l, and this can only be balanced by another couple $= (t \text{ or } c) \times$ A B, a tension being felt in the upper fibres and a compression in the lower ones, shewn respectively by the link A and strut B.

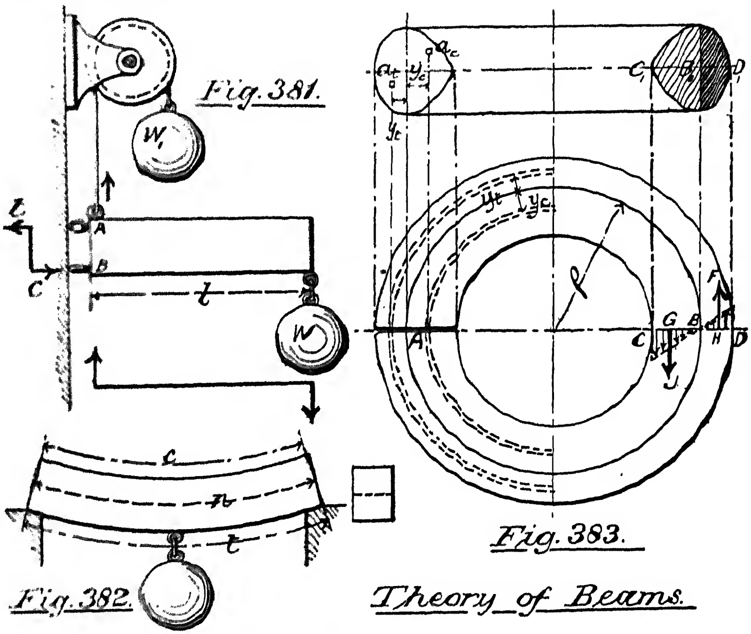


Fig. 381. Fig. 382. Fig. 383.

Theory of Beams.

The case we have examined is that of a *cantilever* or overhung beam. Fig. 382 shews a supported beam or *girder*, and the bending action is here reversed, the lower fibres being in tension and the upper in compression. Taking a bar of indiarubber, and measuring both before and after bending, it will be found that c is thereby shortened, t lengthened, while n is unaltered; n is therefore termed the *neutral line or axis* of the bar.

Position of Neutral Axis.—The bar in Fig. 383 is bent to an entire circle, and has A B for neutral axis, with fibres B C

in compression and BD in tension. The stresses will be zero at B and increase towards C and D as shewn, forming a couple, (J or F) × GH, to resist bending, where J = F. Consider two small areas a_t and a_c, and let ρ = radius of curvature at neutral line. Then:

$$\text{Length before bending} = 2\pi\rho$$
$$\text{Length of ring } a_t \text{ after bending} = 2\pi(\rho + y_t)$$
$$\text{Length of ring } a_c \text{ after bending} = 2\pi(\rho - y_c)$$
$$\therefore \text{ Strain on fibre } a_t = 2\pi(\rho + y_t) - 2\pi\rho = 2\pi y_t$$
$$\text{and Strain on fibre } a_c = 2\pi\rho - 2\pi(\rho - y_c) = 2\pi y_c$$

$$\text{But } \Delta = \frac{fl}{E} \text{ generally} \quad \therefore 2\pi y = \frac{f\,2\pi\rho}{E} \quad \text{and } f = \frac{Ey}{\rho} \ \ldots (1)$$

$$\text{Total stress on a small area} = fa = \frac{Eya}{\rho}$$

$$\text{Total stress on area } B_1D_1 = \text{sum of } \frac{Ey_t a_t}{\rho} \text{ for all portions of } B_1D_1$$

$$\text{and Total stress on area } B_1C_1 = \text{sum of } \frac{Ey_c a_c}{\rho} \text{ for all portions of } B_1C_1$$

But these are the forces F and J,

$$\therefore \Sigma\frac{Ey_t a_t}{\rho} = \Sigma\frac{Ey_c a_c}{\rho} \quad \text{and as } \frac{E}{\rho} \text{ is a constant,}$$

$$\Sigma y_t a_t = \Sigma y_c a_c \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

or Moment of tension area = moment of compression area.

But the centre of gravity of a lamina or centre of figure of area is such that the moments on either side are equal. Therefore *the neutral axis of any bar passes through the centre of figure (or centroid) of its cross section.*

Moment of Resistance.—Again, in Fig. 383.

$$\text{Moment of stress on } a = fa \times y = \frac{Eya}{\rho} \times y$$

$\therefore$ Moment of all stresses on a section

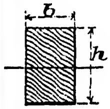
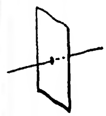
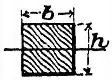
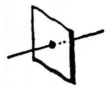
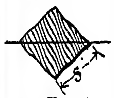
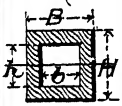
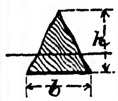
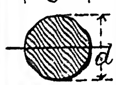
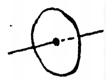
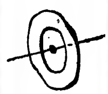
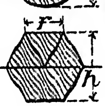
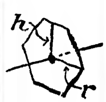
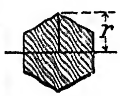
$$= \Sigma\left(\frac{Eya}{\rho} \times y\right) = \frac{E}{\rho}\Sigma ay^2$$

But Σay^2 is the moment of inertia (2nd moment) of the section = I

$$\therefore \text{ Moment of resistance} = \frac{EI}{\rho} \text{ (in terms of } \rho) \ldots\ldots (3)$$

(*See p.* 1078.)

MOMENTS OF INERTIA OF AREA OR 2ND MOMENT (WITH AXIS AS SHEWN).

	For Beams (*rectangular.*)			For Shafts (*polar.*)		
	Section.	I	y		I	y
Rectangle		$\frac{bh^3}{12}$	$\frac{h}{2}$		$\frac{bh(b^2+h^2)}{12}$	$\frac{\sqrt{b^2+h^2}}{2}$
Square		$\frac{bh^3}{12}$	$\frac{h}{2}$		$\frac{s^4}{6}$	$s\sqrt{2}$
Square		$\frac{s^4}{12}$	$\frac{s\sqrt{2}}{2}$	The same general formula holds for shaft moment as for beams, thus $T_m = f\frac{I}{y}$. Note however the restrictions at *p.* 420.		
Hollow Rectangle or Square		$\frac{BH^3 - bh^3}{12}$	$\frac{H}{2}$			
Triangle		$\frac{bh^3}{36}$	$\frac{2}{3}h$			
Circle		$\frac{\pi d^4}{64}$	$\frac{d}{2}$		$\frac{\pi d^4}{32}$	$\frac{d}{2}$
Hollow Circle		$\frac{\pi}{64}(D^4 - d^4)$	$\frac{D}{2}$		$\frac{\pi(D^4 - d^4)}{32}$	$\frac{D}{2}$
Hexagon		$\frac{5\sqrt{3}h^4}{16}$	$\frac{h}{2}$		$\frac{hr^3}{2} + h^3 r$	r
Hexagon			r			

We may also represent the moment in terms of the limiting stress f (sometimes f_c, and sometimes f_t). Then:

Bending moment = moment of resistance

$$\underline{B_m = fZ} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

and Z is known as the modulus of section.*

Let y = distance of *furthest* fibre from axis:

By (1) $f = \frac{Ey}{\rho}$ by (3) $B_m = \frac{EI}{\rho}$ and by (4) $B_m = fZ$

Then $fZ = \frac{EI}{\rho}$ and $\frac{Ey}{\rho} Z = \frac{EI}{\rho}$ $\therefore$ $Z = \frac{I}{y}$ ...... (5)

and $\underline{B_m = f\frac{I}{y}}$ (*See p.* 1079.)

The value of Z can now be found; thus,

for <u>rectangular sections</u> $fZ = f\frac{bh^3}{12} \div \frac{h}{2} = \underline{f\frac{bh^2}{6}}$

and for <u>circular sections</u> $fZ = f\frac{\pi d^4}{64} \div \frac{d}{2} = \underline{f\frac{\pi d^3}{32}}$

Graphic Solution for Moment of Resistance.— Taking, first, a rectangular section (Fig. 384), draw the neutral axis A B. Then C D will be the *line of limiting (or greatest) stress*, and the *value of any horizontal fibre* E F to resist stress will be found by projecting to C D and joining C D to N, thus obtaining the intercept G M. Every fibre being thus treated, the sum of the *virtual stress areas* will be the areas C D N and H J N, which each make one force of the couple when multiplied by the limiting stress f. K and L are the centres of gravity of the areas.

Moment of resistance (generally) = one force × total arm

<u>Moment of rectangular section</u> $= f\begin{Bmatrix}\text{area CDN} \\ \textit{or}\ \text{HJN}\end{Bmatrix} \times \text{arm KL}$

$$= f\left(\frac{bh}{4}\right) \times \frac{2}{3}h = \underline{f\frac{bh^2}{6}}$$

Unsymmetrical sections are treated at Figs. 389, 390 by this method, which can be applied to any section. (*See App. II., p.* 847.)

* Z = virtual area × arm. (See Figs. 384, 385.)

To find the Moment of Inertia of any Beam Section. –Proceed as in the last construction and find Z. Then $Z = \frac{I}{y}$ nd $I = Z\,y$.* So for rectangular sections

$$I = \frac{b\,h^2}{6} \times \frac{h}{2} = \frac{b\,h^3}{12}$$

Stress areas for *circle*, *hollow circle*, *triangle*, and *hollow ectangle* are shewn in Fig. 385, being measured as in Fig. 372 mean width $\times \frac{h}{2}$). The centre of gravity of each area is obtained y cutting out in stiff paper and hanging up in two different

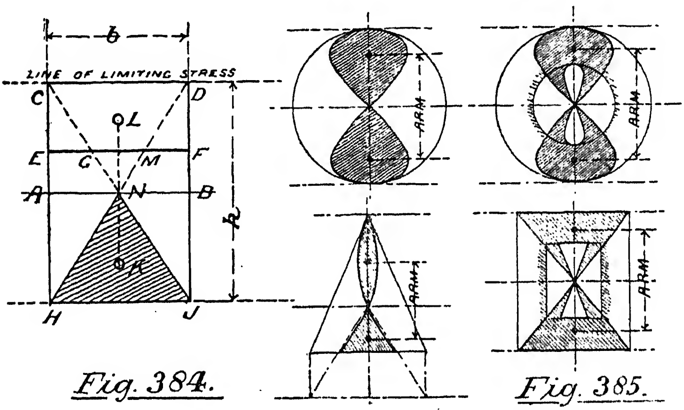


Fig. 384. *Fig. 385.*

ositions to mark two plumb lines, which will cross at G. For he triangular beam the neutral axis must be drawn at $\frac{1}{3}$ the ieight, a line of limiting stress drawn across the apex, and another elow, at an equal distance from its axis. Projecting and con-erging, we shall obtain the areas shewn, which *must be equal.* The results are as follows:

Moment of resistance of circle			$= f \cdot 0982\, d^3$
,,	,,	of hollow circle	$= f \cdot 0982 \frac{D^4 - d^4}{D}$
,,	,,	of triangle	$= f \cdot 0417\, b\,h^2$
,,	,,	of hollow rectangle	$= f \cdot 1666 \frac{B\,H^3 - b\,h^3}{H}$

* See also note, page 430; see also page 845.

Centre of Gravity of Area Centroid by Calculation or Graphics. [illegible] ... then [illegible]

[illegible] — [illegible] lengths [illegible]

[illegible]

[illegible] the centre of gravity. By turning the figure through about 90°, and repeating the process, [illegible] and [illegible] the centre of gravity of the whole figure.

Fig. 386.

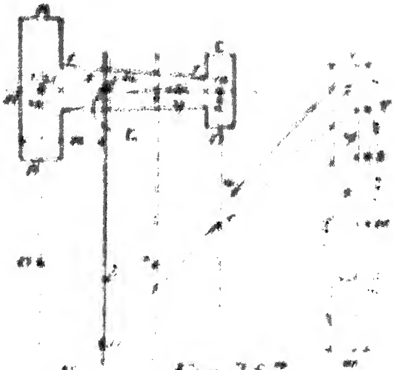
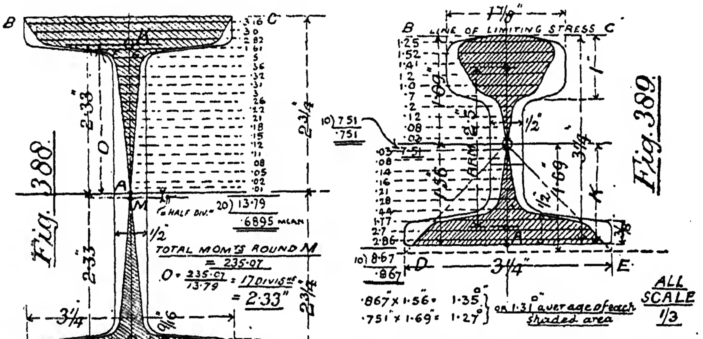


Fig. 387.

The graphic method at Fig. 387 suits some cases better. The area is first divided up, and its separate parts calculated [illegible] and [illegible] being a trapezium is cut into triangles of [illegible] and [illegible]. Next mark centres of each of these areas at [illegible], and [illegible]; set off the weights to scale upon [illegible], join to any point [illegible], and drop perpendiculars through [illegible], and [illegible] of the area. Now commence at any point [illegible] in [illegible] and draw [illegible], and finally [illegible], giving [illegible] which, produced upward, cuts the centre line [illegible] the centre of gravity, and [illegible] is then known. (*See pp.* 843 *and* [illegible].)

Economical Sections.—It will be seen in Fig. 384 that half the material is incapable of resistance on account of its location near the axis, being only affected by shear, which however, has usually but a small effect. We are therefore driven to the conclusion that 'solid' beams are uneconomical (seen also in the solid circle and triangle in Fig. 385). The hollow circle and hollow rectangle are an improvement, but the best results are obtained by distributing the material near the line of limiting stress, and thus the well-known **H** section (Fig. 388) is arrived at for wrought iron, where $f_c = f_t$ approximately, while the modified **T** section (Fig. 390) is adopted for cast iron where $f_c > f_t$.



Moments of Rolled Beams.

Assuming that the vertical web is for the purpose of resisting shear, we may find the moment of resistance by an

Approximate Method.—The direct strength of the flanges forms a couple whose arm may be taken as the depth from centre to centre of the flanges (the vertical web being neglected). Let a = area of total depth of either flange,

$$\text{Moment of resistance of } \mathbf{H} \text{ section} = f_c\, a_c\, h \quad \text{or } f_t\, a_t\, h$$

whichever is the lowest value.

In cast iron $\dfrac{f_t}{f_c} = \dfrac{1\frac{1}{4}}{4}$ or $\frac{1}{4}$ roughly, and the flanges must have areas in inverse proportion.

Exact graphical solution may also be found, and we will take a few cases.

Momental Strength of Wrought Iron Rolled Beam (the section being given at Fig. 388).—Referring every fibre to C B or D E we obtain the shaded stress areas. As these change in contour very abruptly, it is best to divide into 20 parts to find the mean width ·6895. Then ·6895″ × 2·75″ = 1·896 area in sq. ins. The arm may be found by calculation or by hanging up the paper area from two positions, the first method being shewn in the diagram, and the result found as 2·33″ on either side. Then Z = area × arm, and

$$\underline{\text{Moment of resistance}} = f \times 1{\cdot}896 \times 4{\cdot}66 \quad = f\ 8{\cdot}835$$
$$= 4 \times 8{\cdot}835 = \underline{35{\cdot}34 \text{ ton inches}}$$

In such beams $f_t = 5$ tons and $f_c = 4$ tons, so the lowest value has been taken. By the approximate formula,

$$\text{Moment} = f_c a_c h = 4 \times 1{\cdot}625 \times 5 = \underline{32{\cdot}5 \text{ ton inches.}}$$

Momental Strength of Steel Rail (Fig. 389).—By cutting out the section and hanging it, the neutral axis is found at 1·56″ from bottom and 1·69 from top; the limiting line is therefore B C. A second limiting line is drawn at D E, also 1·69 from axis, every fibre now referred to B C or D E, and the stress areas obtained. Cutting these out, their centres can be found, giving the arm 2·5″, and their areas by dividing each into 10 parts vertically. Then (mean width × height) gives ·751″ × 1·69″ = 1·27 for top area, and ·867″ × 1·56″ = 1·35 for bottom area. It is very difficult to get them exactly equal graphically, so the average 1·31 sq. ins. must be taken. Then

$$\underline{\text{Moment of resistance}} = f \times \text{area} \times \text{arm}$$
$$= 6 \times 1{\cdot}31 \times 2{\cdot}5 = \underline{19{\cdot}65 \text{ ton inches.}}$$

Momental Strength of Cast Iron Beam (Fig. 390).—C D A B is the beam section, whose axis is found at E. Draw perpendiculars F G and M N. Set off H G = 1¼ and M K = 4, representing f_t and f_c respectively, and draw H J and K L through axis, giving F J as 2⅝ and L N as 2⅛. This shews that if f_c be the limiting stress the tension flange would be stressed to 2⅛ tons, or dangerously; while f_t at 1¼ tons would only stress the compression flange at 2⅝ tons, or safely. f_t is therefore the limiting stress, and A B the limiting line below E, while a corresponding

one P Q is drawn at equal distance above E. Then as before, after drawing stress areas:

$$\left.\begin{array}{l}\text{Upper area} = 1{\cdot}256 \times 4{\cdot}56 = 5{\cdot}7 \\ \text{Lower area} = 2{\cdot}102 \times 2{\cdot}44 = 5{\cdot}2\end{array}\right\} = 5{\cdot}45 \text{ average}$$

and $\underline{\text{Moment of resistance}} = f_t \times \text{area} \times \text{arm}$

$$= 1\tfrac{1}{4} \times 5{\cdot}45 \times 5{\cdot}35 = \underline{36{\cdot}45 \text{ ton inches}}$$

or by approximate formula $= 1\tfrac{1}{4} \times 6 \times 5\tfrac{1}{8} = \underline{38{\cdot}4 \text{ ton inches.}}$

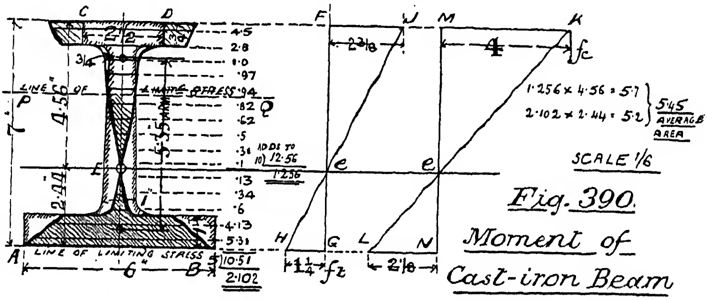
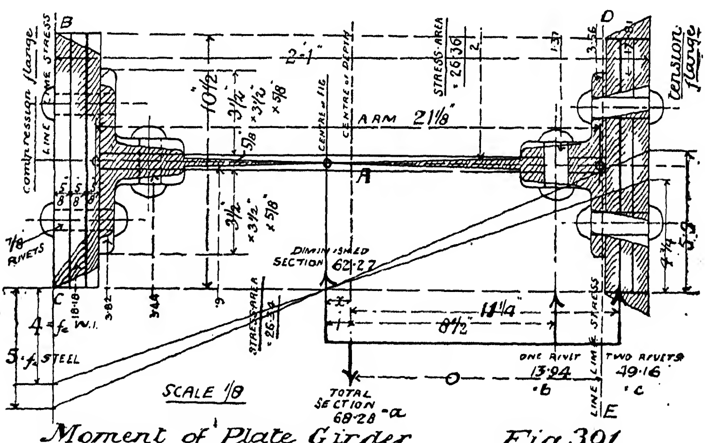


Fig. 390. Moment of Cast-iron Beam

Momental Strength of Plate Girder (Fig. 391).—The rivets must be deducted in tension flange, but they aid the



Moment of Plate Girder. Fig. 391.

resistance in compression. The centre of figure (centroid) is then found by taking moments of the parts round A.

Moment of diminished section = moment of 1 rivet + moment of 2 rivets

$$\{a-(b+c)\}\,x = b \times \text{arm} + c \times \text{arm}$$

$$\text{and } x = \frac{(1{\cdot}394 \times 8{\cdot}5)+(49{\cdot}16 \times 11{\cdot}25)}{68{\cdot}28-(13{\cdot}94+49{\cdot}16)} = 1''$$

f_c being limiting stress (see below), B C and D E are reference lines, and the areas are found as before.

Each stress area = 26·35 and arm = 21·12

For W. I. plate girders $f_t = 5$ and $f_c = 4$

For Steel plate girders $f_t = 6$ and $f_c = 5$

The reduced f_c being an allowance for buckling.

$\therefore$ Moment of resistance $= f_c \times$ area $\times$ arm

$= 4 \times 26{\cdot}35 \times 21{\cdot}12 =$ 2226 ton ins. for W. I.

$= 5 \times 26{\cdot}35 \times 21{\cdot}12 =$ 2782 ton ins. for Steel.

Value of f in Beams.—If a 'solid' beam be broken across, the ultimate stress, deduced by applying the momental formula, will be usually found much greater than f_t breaking. If then, the bending theory be pushed as far as the breaking load, we must meet the case by the value

$$f_o = O\,f_t$$

where f_o is the stress found by transverse experiment and called the *modulus of rupture*, while O we shall call the *bending coefficient*. It varies with the beam section. Thus:

In sections ● or ◆ O is greatest, being about 2

In sections ▮ or ■ O is less, being about $1\frac{1}{2}$

In sections I O = 1

but depends also on the material, as seen in the following table (compiled from experiment), and is often less than unity for woods.

TABLE OF BENDING COEFFICIENTS (O) FOR SOLID SECTIONS.

Fir	▮	·52 to ·94	Wrought Iron	▮ 1·6;	● 1·75
Oak	▮	·7 to 1·0	Forged Steel	▮ 1·47;	● 1·6
Pitch Pine	▮	·8 to 2·2	Gun Metal	▮ 1·0;	● 1·9
Cast Iron	▮ 2;	● 2·35;	I $1+\frac{b}{a}$	where a=flange width and b=web thickness	

And our beam formula becomes $B_m = O\,f\,Z$

In all cases of compound stress there appear to be points of difference between practice and the theories adopted. Thus a complete theory of bending ought to simultaneously consider tension, compression, vertical shear, and horizontal shear, while the bending theory treated on p. 428 deals only with the moment of the direct stresses; hence we should not be surprised to find the differences referred to. Some writers show that longitudinal shear is exceedingly small as compared with the effect of direct stress

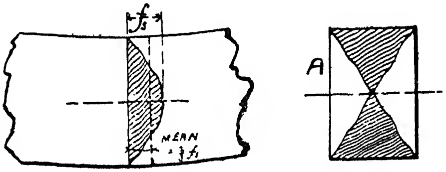


Fig. 393.

(as also is vertical shear, in beams where the ratio of length to breadth is considerable). Others, again, assert they have found theory and practice agree perfectly within the elastic limit of truly elastic material. Without doubt the greatest portion of the discrepancy is due to the attempt to use the beam formula during the plastic stage, for which it was never constructed. Neither can it be correct to proportion beams by using a factor on the modulus of rupture, the method largely adopted up to say 1875 or 1880. Probably a somewhat higher value of f may be used than f_t (but not so high a one as f_0), upon which to use the safety factor in the case of solid beams; while thin built-up sections may be proportioned by putting the safe f_t in the usual formula. The greatest difference between f_t and f_0 occurs, as we should expect, with materials that have no true elastic stage, or, what is the same thing, with such as have a constantly varying value for E. Among these may be mentioned cast iron, india-rubber, and some woods (*see also p.* 453).

We have now completed our investigations of moment of resistance, and shall proceed to consider the left side of the bending equation.

Bending Moment and Vertical Shear.—In long beams the shear is small in comparison with bending stress, and is fully met by the surplus section. For the distribution of shear stress

may be shewn to be parabolic, as at f_s (Fig. 393), or greatest near the axis, while on the contrary the greatest bending stress is furthest from the axis, as shewn at A.

$$\tfrac{2}{3} f_s \times \text{area of section} = \text{total shear}$$

$$\text{and } f_s = \tfrac{3}{2}\ \frac{\text{total shear load on section}}{\text{area of section}}$$

In very short beams this stress should be considered, till finally, in rivets and pins, the shear is almost pure. We will now examine the distribution of Bending Moment and Shear Load under various conditions of support and load. (*See p.* 922.)

I. *Cantilever with Concentrated Load* * (Fig. 394).—A B is the beam and W the load, the latter having a leverage over

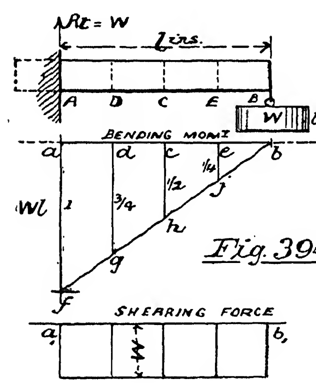


Fig. 394.

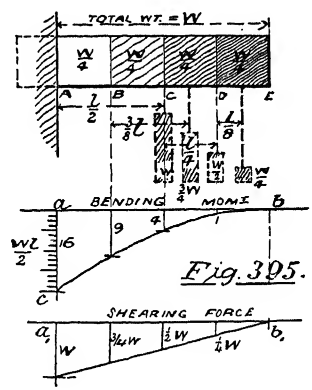


Fig. 395.

section A of $W \times l$ ton feet: at section D of $W \times \frac{3}{4} l$, and so on. The *Bending Moments* at various sections may therefore be represented on the base line *a b* by *downward* ordinates, thus:

$$\text{At A} = W\,l \times 1 \qquad \text{at D} = W\,l \times \tfrac{3}{4}$$
$$\text{At C} = W\,l \times \tfrac{1}{2} \qquad \text{at E} = W\,l \times \tfrac{1}{4}$$

and at B = nothing; and these ordinates are shewn at *f*, *g*, *h*, *j*, and *b*.

The *Shearing Force* is caused by the reciprocal action of W and R_t, and will equal W upon any section between A and B. For regularity we shall always consider the force *on the right side of*

* Weight of beam is not taken unless stated.

the section only, so here the shear ordinates are drawn *downward* on the line $a\ b$, and equal W in every case.

II. *Cantilever with Uniformly Distributed Load* (Fig. 395), the load being represented by the weight of the beam. Considering the beam hinged successively at A, B, C, D, and E, the loads on the right may be successively concentrated at their middle points, and the *Bending Moments* become:

$$\text{At A} = W \times \frac{l}{2} = W\,l \times \frac{1}{2} \quad \propto 16 = 4^2$$

$$\text{At B} = \frac{3}{4} W \times \frac{3}{8} l = W\,l \times \frac{9}{32} \quad \propto 9 = 3^2$$

$$\text{At C} = \frac{W}{2} \times \frac{l}{4} = W\,l \times \frac{1}{8} \quad \propto 4 = 2^2$$

$$\text{At D} = \frac{W}{4} \times \frac{l}{8} = W\,l \times \frac{1}{32} \quad \propto 1 = 1^2$$

and at E = nothing, as shewn by diagram, and as the ordinates vary as the square of the abscissæ at $a\ b$, the curve is a parabola with b as vertex.

Shearing Force on right of section A = W, at section B $= \frac{3}{4}$ W, and at C D and E, $\frac{W}{2}$, $\frac{W}{4}$, and nothing respectively, as shewn by diagram on $a_1\,b_1$.

III. *Girder with concentrated load at centre and ends merely supported* (Fig. 396).—Reactions will each equal $\frac{W}{2}$, which we shall use in estimating the moment. R_{t_1} balances R_{t_2} round W as a pivot, and the stress at E is due to *one or the other*, but not to both. Then calling each reaction $\frac{W}{2}$ the *Bending Moments* will be:

$$\text{At B} = \frac{W}{2} \times \frac{l}{8} = W\,l \times \frac{1}{16} \quad \propto 1$$

$$\text{At C} = \frac{W}{2} \times \frac{l}{4} = W\,l \times \frac{1}{8} \quad \propto 2$$

$$\text{At D} = \frac{W}{2} \times \frac{3}{8} l = W\,l \times \frac{3}{16} \quad \propto 3$$

$$\text{At E} = \frac{W}{2} \times \frac{l}{2} = W\,l \times \frac{1}{4} \quad \propto 4$$

and similarly between J and E.

[illegible]

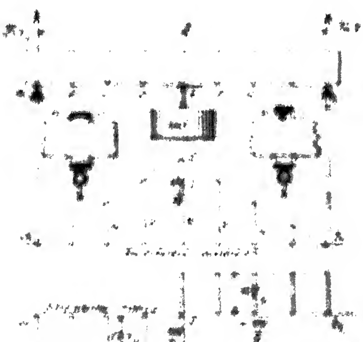


Fig. 396

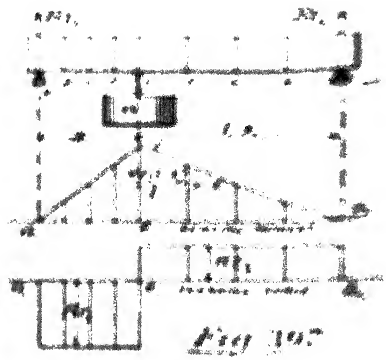


Fig. 397

IV. *Girder with concentrated load at any point.* Taking moments round 1, we have

$$R_2 \times l = W(l - x) \quad \text{and} \quad R_1 = \frac{W x}{l}$$

Bending Moment at $x = \frac{W(l - x)}{l} x$, and the rest of the diagram is made up of straight lines [illegible] as in the last case. The *Shearing Force* is found from the reactions as before.

V. *Girder supported at ends and loaded uniformly* (Fig. 397), the weight of the beam representing the load. Reactions $= \frac{W}{2}$ in each case. The *Bending Moments* are the subtraction of the weight moment and reaction moment at every 'hinge,' as shown diagrammatically.

$$\text{At B} = \left(\frac{W}{2} \times \frac{l}{8}\right) - \left(\frac{W}{8} \times \frac{l}{16}\right) = W\,l \times \frac{7}{128} \quad \propto \quad 7$$

$$\text{At C} = \left(\frac{W}{2} \times \frac{l}{4}\right) - \left(\frac{W}{4} \times \frac{l}{8}\right) = W\,l \times \frac{3}{32} \quad \propto \quad 12$$

$$\text{At D} = \left(\frac{W}{2} \times \frac{3}{8}l\right) - \left(\frac{3}{8}W \times \frac{3}{16}l\right) = W\,l \times \frac{15}{128} \quad \propto \quad 15$$

$$\text{At E} = \left(\frac{W}{2} \times \frac{l}{2}\right) - \left(\frac{W}{2} \times \frac{l}{4}\right) = W\,l \times \frac{1}{8} \quad \propto \quad 16$$

and are similarly found between H and E, all being shewn on base $a\,b$. Drawing $f\,k$ horizontally, the intercepts between $k\,f$ and $a\,f$ are seen to vary as in Fig. 395, and the curve is therefore a parabola with vertex at f.

The *Shearing Force* is found by deducting upward and downward forces on the right of each section. Thus:

$$\text{At A} = W - \frac{W}{2} = W \times \frac{1}{2} \quad \propto \quad 4$$

$$\text{At B} = \frac{7}{8}W - \frac{W}{2} = W \times \frac{3}{8} \quad \propto \quad 3$$

$$\text{At C} = \frac{3}{4}W - \frac{W}{2} = W \times \frac{1}{4} \quad \propto \quad 2$$

$$\text{At D} = \frac{5}{8}W - \frac{W}{2} = W \times \frac{1}{8} \quad \propto \quad 1$$

$$\text{At E} = \frac{W}{2} - \frac{W}{2} = 0 \quad \propto \quad 0$$

$$\text{At F} = \frac{3}{8}W - \frac{W}{2} = -W \times \frac{1}{8} \quad \propto -1$$

$$\text{At G} = \frac{W}{4} - \frac{W}{2} = -W \times \frac{1}{4} \quad \propto -2$$

$$\text{At H} = \frac{W}{8} - \frac{W}{2} = -W \times \frac{3}{8} \quad \propto -3$$

$$\text{At J} = -\frac{W}{2} = -W \times \frac{1}{2} \quad \propto -4$$

There is no force whatever at the centre.

VI. *Beam fixed at both ends and loaded in the centre* (Fig. 399), weight of beam neglected. The beam will be deflected to the dotted shape, A C and G J acting as cantilevers, and C G as a supported girder. From C to G the B_m is upward, and from

[illegible] upwards, being zero at c and c_1, the points of contraflexure in this case at [illegible] from the ends. The *Bending Moments* are:

$$B \text{ at } c = \left(\frac{W_c}{4}\right) \text{ generally} = [illegible] - Wl + [illegible] \text{ upward}$$

$$B \text{ at } [illegible] = [illegible] \text{ generally} = [illegible] - Wl + [illegible] \text{ downward}$$

and the diagrams are given in [illegible].

Shearing Force is the same as Case III. See [illegible].

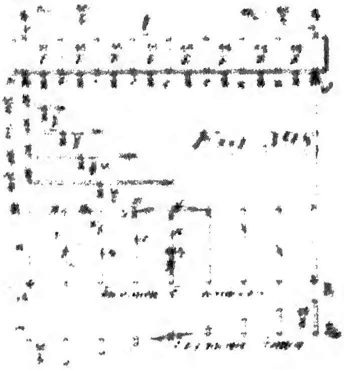


Fig. 308.

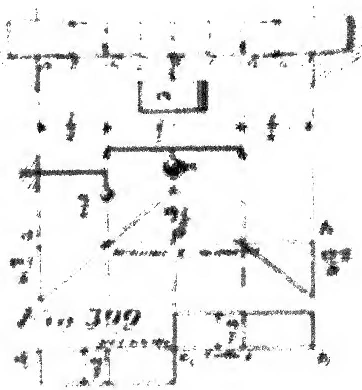


Fig. 309.

VII. *Beam fixed at both ends and loaded uniformly by its own weight* (Fig. 309). The points of contraflexure, c and c_1, are found to be ·211 l from either end. Then *Bending Moment*

$$\text{at } c = \left(\frac{Wl}{8}\right) \text{ generally} = \frac{·578\,W \times ·578\,l}{8} = ·04176\,Wl \text{ or } \frac{1}{24}Wl,$$

and cc_1 is a parabola drawn upwards.

The moments on ac are compounded thus:

For concentrated load ·289 W, B_a = ·289 W × ·211 l = ·06098 W l

and for uniform load ·211 W, $B_a = \dfrac{·211\,W \times ·211\,l}{2}$ = ·02226 W l

These added give ·0833 W l or $\dfrac{Wl}{12}$.

Drawing [illegible] on base [illegible], and the parabola [illegible] on base [illegible] [illegible]

Case II.), it will be found that the total ordinates under $p\,q$ vary as the square of their distance from q, proving that $k\,q\,l_1$ is a continuous parabola.

The *Shearing Force* at A consists of

$$\cdot 289\ \mathrm{W} + \cdot 211\ \mathrm{W} = \frac{\mathrm{W}}{2} \quad \text{and at C} = \cdot 289\ \mathrm{W}$$

and the diagram is a straight line. *(See Appendix II., p.* 848.)

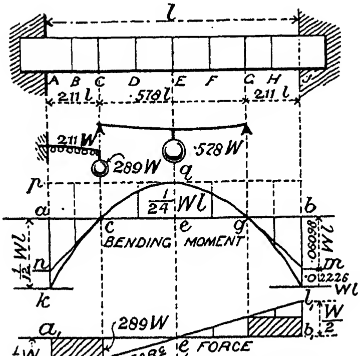


Fig. 400.

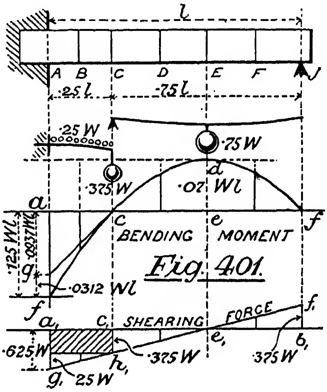


Fig. 401.

VIII. *Beam fixed at one end, supported at the other, and loaded by its own weight* (Fig. 401).—This case may be deduced from the first figure on p. 850, and the point of contra-flexure is ·25 l from A and ·75 l from J. *Bending Moment*

$$\text{At E} = \left(\frac{\mathrm{W}\,l}{8}\right) \text{generally} = \frac{\cdot 75\ \mathrm{W} \times \cdot 75\ l}{8} = \cdot 07\ \mathrm{W}\,l$$

$$\left.\begin{aligned} &\text{At } a\,g = \cdot 375\ \mathrm{W} \times \cdot 25\ l = \cdot 0937\ \mathrm{W}\,l \\ \text{and}\quad &\text{At } g\,f = \frac{\cdot 25\ \mathrm{W} \times \cdot 25\ l}{2} = \cdot 0312\ \mathrm{W}\,l \end{aligned}\right\} = \cdot 125\ \mathrm{W}\,l$$

Shearing Force at $\mathrm{A} = a_1\,d_1 + d_1\,g_1 = \cdot 625$ W, and at $\mathrm{J} = b_1\,f_1 = \cdot 375$ W. The curves will be found continuous in both diagrams. *(See Appendix II., p.* 852.)

Combination Diagrams are shewn in Fig. 402 based on cases already discussed. The final shaded areas $\mathrm{B_f}$ and $\mathrm{S_f}$ are found

by superposing the results due to the separate loads, having regard to the signs + and −. The cases are as follows:

(1.) One distributed and two concentrated loads on cantilever.
(2.) One concentrated and one distributed load on girder.
(3.) Maximum diagram for rolling load.
(4.) Three concentrated loads and one distributed load on girder.

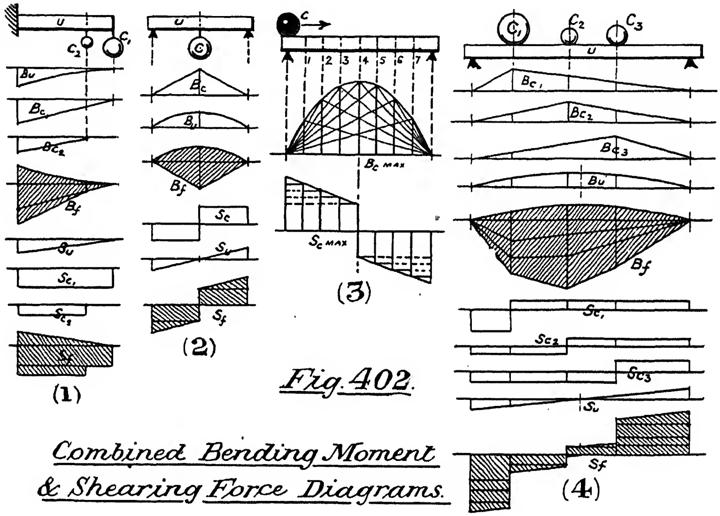


Fig. 402.

Combined Bending Moment & Shearing Force Diagrams.

The distributed load is more conveniently placed on one side of the base line, and the concentrated loads on the other side, in the superposed diagram. All are well lettered to shew the relation between diagram and load. In Case (3) the load must be placed over the successive numbers, and diagrams obtained for every position, as in Fig. 397, then the bounding curve will be the maximum, and the final maximum B_m curve will be a parabola.

Fig. 403 shews a continuous beam on three supports, loaded by equal concentrated loads, W_1 and W_1, and uniformly by its own weight, W + W. The contra-flexure points are practically the same for each case, and the diagrams can be obtained from previous considerations (*see p.* 953). The following table is very useful for continuous beams:—

TABLE OF REACTIONS ON SUPPORTS FOR CONTINUOUS BEAMS, AS FOUND FROM CLAPEYRON'S 'THEOREM OF THREE MOMENTS' IN TERMS OF W, THE UNIFORM LOAD ON EACH SPAN.

(*See also Appendices I. and II. pp.* 762 *and* 849.)

2 Spans	$\frac{3}{8}$	$\frac{10}{8}$	$\frac{3}{8}$							
3 Spans	$\frac{4}{10}$	$\frac{11}{10}$	$\frac{11}{10}$	$\frac{4}{10}$						
4 Spans	$\frac{11}{28}$	$\frac{32}{28}$	$\frac{26}{28}$	$\frac{32}{28}$	$\frac{11}{28}$					
5 Spans	$\frac{15}{38}$	$\frac{43}{38}$	$\frac{37}{38}$	$\frac{37}{38}$	$\frac{43}{38}$	$\frac{15}{38}$				
6 Spans	$\frac{41}{104}$	$\frac{118}{104}$	$\frac{108}{104}$	$\frac{106}{104}$	$\frac{108}{104}$	$\frac{118}{104}$	$\frac{41}{104}$			
7 Spans	$\frac{56}{142}$	$\frac{161}{142}$	$\frac{137}{142}$	$\frac{143}{142}$	$\frac{143}{142}$	$\frac{137}{142}$	$\frac{161}{142}$	$\frac{56}{142}$		
8 Spans	$\frac{152}{388}$	$\frac{440}{388}$	$\frac{374}{388}$	$\frac{392}{388}$	$\frac{386}{388}$	$\frac{392}{388}$	$\frac{374}{388}$	$\frac{440}{388}$	$\frac{152}{388}$	
9 Spans	$\frac{209}{530}$	$\frac{601}{530}$	$\frac{511}{530}$	$\frac{535}{530}$	$\frac{529}{530}$	$\frac{529}{530}$	$\frac{535}{530}$	$\frac{511}{530}$	$\frac{601}{530}$	$\frac{209}{530}$

Culmann's Funicular Polygon (Fig. 404) is a ready means of solving such a problem as (4) Fig. 402. Culmann of Zurich proved that the bending moments are there proportional to the ordinates of a polygon obtained by hanging the same weights to a loose string hooked at the supports. Taking the loads in Fig. 404, B C is drawn to scale, and represents the weights taken in order shewn. Mark a point E any distance x ft. from C B, and join to C, M, N, and B. Draw from any point F, F G $\parallel$ C E, G H $\parallel$ M E, H J $\parallel$ N E, and J K $\parallel$ B E. Join K F, and draw E L $\parallel$ K F. The shaded polygon is the curve of *Bending Moment*, and

$$\underline{B_m = \text{vertical ordinate in lbs.} \times x'} \text{ (lb. ft.)}$$

Project S T from L, U V from C, W X from M, Y Z from N, and $a\,b$ from B, and the curve of *Shearing Force* is obtained, measureable by load scale. Also $S\,a = R_t$ and $T\,V = R_{t_2}$ (*See pp.* 855 *and* 1085.)

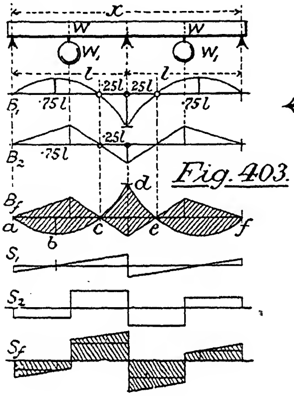


Fig. 403.

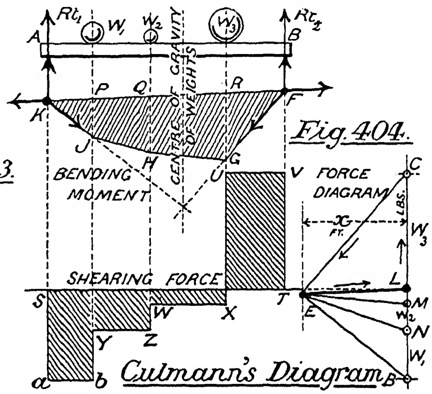


Fig. 404.

Culmann's Diagram.

A Parabola may be drawn by the method in Fig. 405, which consists in dividing A B and B C into an equal number of parts, and joining the divisions of A B to D by lines cutting the divisions of B C, then tracing the curve through the crossing points.

We will now take some examples to illustrate the equation of B_m to $f Z$. The shear diagram is not often required in practice, but should at least be made for trial in short beams.

Example 22.—The following beams are proposed for a central load and given span: (1) a bar 4″ deep by 2″ wide; (2) a bar 3·8″ dia.; (3) a bar 3·5″ square. What are their relative strengths? (Eng. Exam. 1885.)

$$B_m = f Z \quad \text{or} \quad \frac{W\,l}{4} = f Z \qquad \therefore\ W \propto Z$$

$$(1) \qquad Z = \frac{b\,h^2}{6} = \frac{2 \times 16}{6} = \underline{5{\cdot}33 \quad \propto 1}$$

$$(2) \qquad Z = \frac{\pi\,d^3}{32} = \frac{22 \times 3{\cdot}8 \times 3{\cdot}8 \times 3{\cdot}8}{7 \times 32} = \underline{5{\cdot}388 \quad \propto 1{\cdot}01}$$

$$(3) \qquad Z = \frac{s^3}{6} = \frac{3{\cdot}5 \times 3{\cdot}5 \times 3{\cdot}5}{6} = \underline{7{\cdot}145 \quad \propto 1{\cdot}34}$$

Example 23.—(1) A beam 2′ long × 1″ square is broken by 250 lbs. at the centre. (2) Find the breaking load for a beam 10 ft. long, 10″ deep, and 6″ wide, with the load 2 ft. from one end. (Eng. Exam. 1886.)

For (1) (keeping l in feet and $b\,h$ in ins.)

$$\therefore \frac{W\,l}{4}=f\frac{b\,h^2}{6} \quad \therefore f=\frac{W\,l\,6}{4\,b\,h^2}=\frac{250\times 2\times 6}{4\times 1\times 1}=750$$

For (2) by Case IV., Fig. 397

$$B_m=\frac{W\,(l-x)}{l}\,x=\frac{W\;8\times 2}{10}=\frac{8}{5}\,W$$

$$\therefore \frac{8}{5}\,W=f\frac{b\,h^2}{6} \quad \text{and } \underline{W}=f\frac{b\,h^2\times 5}{6\times 8}=\frac{750\times 6\times 100\times 5}{6\times 8}$$

$$=46875 \text{ lbs.}=\underline{20{\cdot}9 \text{ tons.}}$$

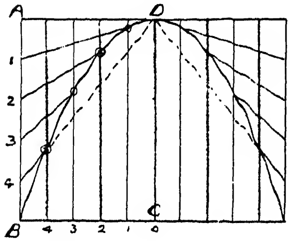


Fig. 405.

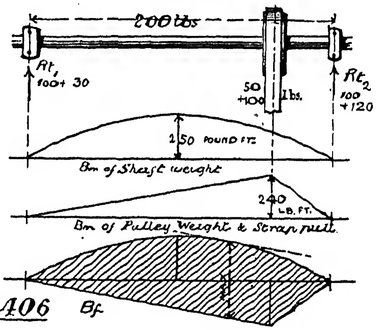


Fig 406

Example 24.—A shaft pulley is 8′ from one bearing, and 2′ from the other. Weight of shaft = 200 lbs. Weight of pulley = 50 lbs. Total belt tension (downwards) = 100 lbs. Draw bending moment diagram and find loads on bearings. (Eng. Exam. 1888.)

$$\underline{B_m \text{ of shaft weight}}=\frac{W\,l}{8}=\frac{200\times 10}{8}=\underline{250 \text{ lb. ft.}}$$

$$\text{Reactions due to pulley and strap}=\frac{2}{10}\times 150=\;\;30 \text{ lbs.}$$

$$\text{and } \frac{8}{10}\times 150=120 \text{ lbs.}$$

$$\therefore\; \underline{B_m \text{ due to pulley weight and strap pull}}=30\times 8 \text{ or } 120\times 2=\underline{240 \text{ lb. ft.}}$$

$$\underline{R_{t1}}=100+\;\;30=\underline{130 \text{ lbs.}}$$

$$\underline{R_{t2}}=100+120=\underline{220 \text{ lbs.}}$$

And the diagrams are shewn at Fig. 406, B_f being the combined figure.

Example 25.—Find the safe concentrated load in the following cases by the approximate formula.

(1) *Wrought Iron Plate Girder.*—Each flange, $10'' \times \frac{1}{2}''$; angles, $3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{1}{2}''$; total depth, 3 ft.; span, 28 ft.; f_t or $f_c = 5$ tons. Allow for $\frac{3}{4}''$ rivets.

(2) *Wrought Iron Rolled Cantilever*, H Section.—Each flange, $4\frac{1}{2}'' \times \frac{5}{8}''$; depth, 8''; overhang, 8 ft.; f_t or $f_c = 5$ tons.

(3) *Cast Iron Girder.*—One flange, $3'' \times 1\frac{1}{2}''$; one flange, $9'' \times 1\frac{1}{2}''$; depth, 12''; span, 20 feet.

(All from Eng. Exam. 1891 and 1892.)

$$(1)\quad W = \frac{4fah}{l} = \frac{4 \times 5 \times (8\frac{1}{2} \times \frac{1}{2} + 12\frac{3}{4} \times \frac{1}{2}) \times 35}{28 \times 12} = \underline{22 \cdot 13 \text{ tons}}$$

$$(2)\quad W = \frac{fah}{l} = \frac{5 \times 4\frac{1}{2} \times \frac{5}{8} \times 7\frac{3}{8}}{8 \times 12} = \underline{1 \cdot 08 \text{ tons}}$$

$$(3)\ f_c\, a_c = 3 \times 1\frac{1}{2} \times 4 = 18 \text{ tons}; \text{ and } f_t\, a_t = 9 \times 1\frac{1}{2} \times 1\frac{1}{4} = 17 \text{ tons}$$

$$\therefore\quad W = \frac{4 \times 17\, h}{l} = \frac{4 \times 17 \times 10\frac{1}{2}}{20 \times 12} \text{ tons} = \underline{2 \cdot 97 \text{ tons.}}$$

Example 26.—Find the depth of an engine guide bar 10'' wide and 4 ft. span. Total piston pressure = 25 tons; length of connecting rod = twice stroke; and greatest obliquity supposed to occur with guide block at centre of span. $f_0 = 5$ tons.

(Hons. Mach. Constr. Ex. 1892.)

Draw crank and rod to scale, Fig. 407. Then the forces are as at A, and the triangle of forces is drawn parallel, as at D.

But $\dfrac{E}{D} = \dfrac{\text{press. on bars}}{\text{piston press.}}$ and press. on bars $= \dfrac{1 \times 25}{4} = 6 \cdot 25$ tons

Then $\dfrac{Wl}{4} = f_0 \dfrac{bh^2}{6}$ and $\underline{h} = \sqrt{\dfrac{Wl \times 6}{4 \times bf_0}} = \sqrt{\dfrac{6 \cdot 25 \times 4 \times 12 \times 6}{4 \times 10 \times 5}} = \underline{3''}$

Example 27.—The girder stays of a combustion chamber are 21 span, and are spaced $8\frac{1}{4}''$ centres apart (see Figs. 311 and 312), the section being rectangular, $5\frac{1}{2}''$ deep by $1\frac{5}{8}''$ wide. There are two bolts to each stay, 7'' apart (Fig. 408). Find the greatest stress in the stay when steam pressure = 225 lbs. per sq. in. (Hons. Mach. Constr. Ex. 1891.)

The roof plate is equivalent to a continuous girder over three spans, as at p. 445, so the pull on each bolt would apparently be $\frac{11}{10}$ W. But the plate is continuous transversely also. Therefore

Each bolt supports $\frac{11}{10}$ of $\frac{11}{10}$ W $= 1{\cdot}21$ W
$= 1{\cdot}21 \times 7 \times 8\tfrac{1}{4} \times 225$ lbs. $= 7$ tons.

Max. B_m at B_1 or $B_2 = R_t \times 7'' = \dfrac{2}{3} \times 7 \times 7 = 32\tfrac{2}{3}$ ton ins.

and Max. B_m at $B_f = 32\tfrac{2}{3} + \dfrac{32\tfrac{2}{3}}{2} = 49$ ton ins.

$$B_m = f\frac{bh^2}{6} \quad \text{and } f = \frac{49 \times 6}{1\tfrac{5}{8} \times 5\tfrac{1}{2} \times 5\tfrac{1}{2}} = \underline{6 \text{ tons per sq. in.}}$$

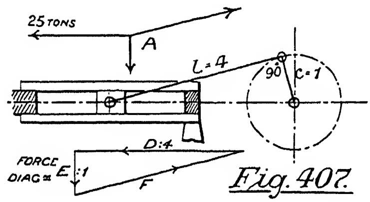


Fig. 407.

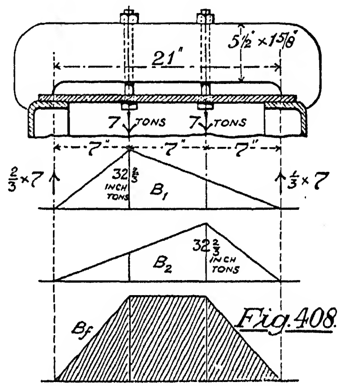


Fig. 408.

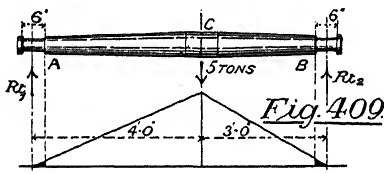


Fig. 409.

Example 28.—An axle is loaded as in Fig. 409, with 5 tons at C. Find greatest B_m; B_π tending to fracture each journal; and deduce the diameters at these places, taking $f_o = 5$ tons. (Hons. Mach. Constr. Ex. 1879.)

$$R_{t1} = \frac{3}{7} \times 5 = 2{\cdot}14 \text{ tons} \qquad R_{t2} = \frac{4}{7} \times 5 = 2{\cdot}85 \text{ tons}$$

$$B_m \text{ at } C = 2{\cdot}14 \times 4 \times 12 = \underline{102{\cdot}72 \text{ ton inches}}$$

$$\therefore \quad B_m \text{ at } A = \frac{3}{48} \times 102{\cdot}72 = \underline{6{\cdot}42 \text{ ton inches}}$$

$$B_m \text{ at } B = \frac{3}{36} \times 102{\cdot}72 = \underline{8{\cdot}56 \text{ ton inches}}$$

$$\left.\begin{aligned} B_m &= f_o\frac{\pi d^3}{32} \\ \text{and} \quad d &= \sqrt[3]{\frac{B_m\, 32}{f_o\, \pi}} \end{aligned}\right\} \quad \begin{aligned} &\therefore \ \underline{d\text{ (at B)}} = \sqrt[3]{\frac{8{\cdot}56 \times 32 \times 7}{5 \times 22}} = \underline{2{\cdot}6''} \\ &\text{and } \underline{d\text{ (at A)}} = \sqrt[3]{\frac{6{\cdot}42 \times 32 \times 7}{5 \times 22}} = \underline{2{\cdot}35''} \end{aligned}$$

Now in circular beams $\frac{4}{10}$ area goes for bending, and $\frac{6}{10}$ goes for shear (see Fig. 385, p. 431).

Total shear at A = f_s × area left for shear
[illegible]

Average f_s = [illegible], and by [illegible] max f_s
[illegible] tons

Similarly, max f_s at B = [illegible] tons, so both journals are safe for shear.

Finally, d at [illegible] = $\sqrt{[illegible]}$ = [illegible]

Deflection of Beams. Take a supported girder, as in

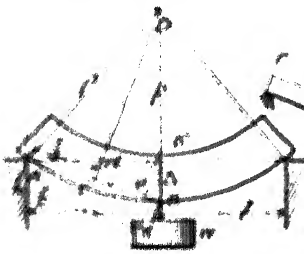


Fig 410.

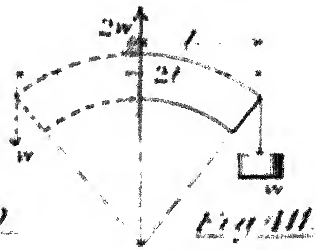


Fig 411.

Fig. 410, of uniform section, and imagine it deflected into an arc of radius ρ. Bisect A B at C. Then B C E is similar to A σ B, because θ is common, while σ and α are right angles. If Δ is small,

$$\frac{\rho}{\frac{l}{4}} = \frac{\frac{l}{2}}{\Delta} \quad \text{and } \Delta = \frac{l^2}{8\rho}$$

$$\text{But } B_m = \frac{E\,I}{\rho} \quad \text{and } \rho = \frac{E\,I}{B_m}$$

$$\therefore \Delta = \frac{l^2 B_m}{8\,E\,I} = \frac{w\,l}{4} \times \frac{l^2}{8\,E\,I} = \frac{w\,l^3}{32\,E\,I}$$

In reality the arc would not be circular but similar to F G, and the deflection would be less; then,

$$\Delta = \frac{w\,l^3}{48\,E\,I}$$ for a girder of uniform section, with central load,

and this will hold if the elastic limit be not exceeded.

Taking two cantilevers back to back, as in Fig. 411, we must substitute for w and l in the above formula $2w$ and $2l$ respectively; then,

For cantilever with concentrated load $\Big\}$ $\Delta = \frac{2w \times (2l)^3}{48\,E\,I} = 16 \times \frac{w\,l^3}{48\,E\,I}$

DEFLECTION FOR UNIFORM BEAMS, WHEN $Y = \dfrac{w\,l^3}{48\,E\,I}$.

Cantilever with concentrated load	16 Y
Cantilever with distributed load	6 Y
Girder with concentrated load	1 Y
Girder with distributed load	$\frac{5}{8}$ Y
Fixed beam with concentrated load	$\frac{1}{5}$ Y
Fixed beam with distributed load	$\frac{1}{8}$ Y
Beam supported one end, fixed at other, central load	$\frac{4}{9}$ Y
Beam supported one end, fixed at other, distributed load	$\frac{4}{15}$ Y

$$\therefore \quad \Delta \propto \frac{w\,l^3}{b\,h^3} \quad \text{and stiffness} \propto \frac{b\,h^3}{l^3}$$

and the practicable allowable deflection is, for cantilevers $\frac{1}{80}''$, and for girders $\frac{1}{40}''$ per ft. of span. (*See Appendix II., p.* 855.)

The **Resilience of a Beam** is equal to half the proof or elastic load multiplied by the corresponding deflection (see p. 367). For a girder with central load,

$$\Delta = \frac{w\,l^3}{48\,E\,I} \qquad w = \frac{4\,f^{\text{lbs}}\,b\,h^2}{6\,l} \qquad \text{and } I = \frac{b\,h^3}{12}$$

$$\therefore \underline{\text{Resilience}} = \frac{w}{2} \times \Delta = \frac{w}{2} \times \frac{w\,l^3}{48\,E\,I} = \frac{16\,f^{2}_{\text{lbs}}\,b^2\,h^4\,l^3\,12}{36\,l^2\,96\,E\,b\,h^3} = \underline{\frac{f^{2}_{\text{lbs}}\,b\,h\,l}{18\,E}} \quad \text{and} \propto b\,h\,l$$

The Strength of Flat Surfaces in Boilers is best calculated by the Board of Trade empirical rule.

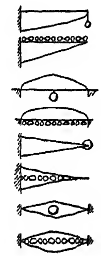
$$\text{Safe steam pressure } p = \frac{C\,(t+1)^2}{s-6}$$

where s = surface supported by one stay, in sq. ins.
t = plate thickness.

$$C = \begin{cases} 100 \text{ when stays have nuts and large washers.} \\ 60 \text{ ditto, but exposed to flame.} \\ 36 \text{ stays riveted over and exposed to flame.} \end{cases}$$

Beams of Uniform Strength.—If rectangular beams be proportioned to their bending moment at every section, the depth or width will vary as follows, easily proved by equation :—

With constant breadth	Case I.	Depth	$\propto$ parabola.
	" II.	"	$\propto$ triangle.
	" III., IV.	"	$\propto$ two parabolas.
	" V.	"	$\propto$ semi-ellipse.
With constant depth	Case I.	Breadth	$\propto$ triangle.
	" II.	"	$\propto$ 2 convex parabolas.
	" III., IV.	"	$\propto$ 2 triangles.
	" V.	"	$\propto$ 2 concave parabolas.



Example 29.—A beam of oak, supported at the ends, 2′ long, 2′ broad, 2″ deep, supports 400 lbs. safely, at the centre, and its deflection is ·06″. Find safe load at centre, and deflection of a beam of oak 16′ long, 9″ broad, 14″ deep, (1) with ends supported; (2) with ends fixed. (Eng. Exam. 1882.)

Taking l in feet and $b\,h$ in ins. $W \propto \dfrac{b\,h^2}{l}$ and $\Delta \propto \dfrac{W\,l^3}{b\,h^3}$

$$\text{Sample beam, } W \propto \frac{2\times 2^2}{2} = 4 \qquad \Delta \propto \frac{400\times 8}{2\times 8} = 200$$

$$\text{New beam, } W_1 \propto \frac{9\times 14\times 14}{16} = 110{\cdot}2$$

(1) Supported; $W : W_1 :: 4 : 110{\cdot}2$ and $\underline{W_1 = \dfrac{400\times 110{\cdot}2}{4} = 11{,}020 \text{ lbs.}}$

$$\Delta_1 \propto \frac{11{,}020\times 4096}{9\times 2744} = 25{,}600 \left\{ \begin{array}{l} \therefore\ \Delta : \Delta_1 :: 200 : 1828 \\ \text{and } \underline{\Delta_1 = \dfrac{1828\times{\cdot}06}{200} = {\cdot}548''} \end{array} \right.$$

(2) Fixed; $B_m(1) : B_m(2) :: \dfrac{W\,l}{4} : \dfrac{W\,l}{8}$ and $\Delta_2 = \frac{1}{5}\,\Delta_1$

$$\therefore\ \underline{W_2 = 11{,}020\times 2 = 22{,}040 \text{ lbs.}} \qquad \therefore\ \underline{\Delta_2 = \frac{{\cdot}548}{5} = {\cdot}109''}$$

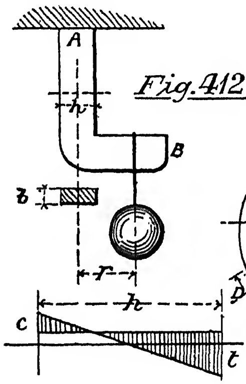
Example 30.—A beam of uniform section is supported at the ends and loaded centrally. Find the ratio of depth to span that the deflection may not exceed $\frac{1}{1000}$ of span when $f = 8000$ lbs. and $E = 28{,}000{,}000$. (Hons. Mach. Constr. Ex. 1887.)

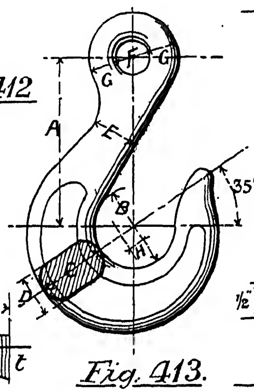
$$\Delta = \frac{w\,l^3}{48\,\mathrm{E\,I}} \quad \text{and as} \quad \frac{w\,l}{4} = f^{\mathrm{lbs}}\frac{2\,\mathrm{I}}{h}, \quad w = \frac{8\,f^{\mathrm{lbs}}\,\mathrm{I}}{h\,l}$$

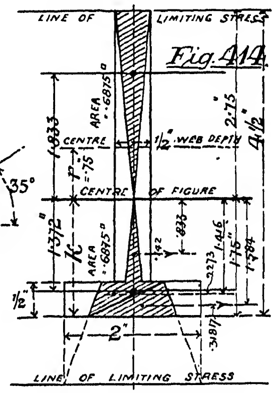
$$\therefore \quad \Delta = \frac{8\,f\,\mathrm{I}\,l^3}{48\,h\,l\,\mathrm{E\,I}} = \frac{f\,l^2}{6\,h\,\mathrm{E}} \quad \text{and} \quad \frac{l}{1000} = \frac{8000 \times l^2}{6 \times 28{,}000{,}000\,h}$$

$$\therefore \quad \frac{l}{h} = \frac{6 \times 28{,}000{,}000}{8000 \times 1000} = \frac{21}{1}$$

Combined Bending and Tension Stress-Action.—Let the bracket in Fig. 412 support a weight W. There are two







actions upon the section: bending due to moment Wr, and tension by direct load W. Then—

(1) Bending action: $\mathrm{W}\,r = f_0\,\mathrm{Z}$ and $f_0 = \dfrac{\mathrm{W}\,r}{\mathrm{Z}}$

or $f_t\,\mathrm{O} = \dfrac{\mathrm{W}r}{\mathrm{Z}}$ and $f_t = \dfrac{\mathrm{W}\,r}{\mathrm{Z\,O}}$

(2) Tensile action: $\mathrm{W} = f_t\,a$ and $f_t = \dfrac{\mathrm{W}}{a}$

$\therefore$ Maximum tensile stress (on inner edge of bh) $= \mathrm{F}_t = \dfrac{\mathrm{W}}{a} + \dfrac{\mathrm{W}r}{\mathrm{Z\,O}}$

O = 1 for **H** and **T** sections, and O = 1½ or 2 for solid sections. (*See p.* 436.)

Strength of Crane Hooks.—In these, theory and practice are considerably at variance. The following table is regularly

used at Elswick, and has been well tested, the diagram being given at Fig. 413.

WROUGHT IRON CRANE HOOKS (ELSWICK PRACTICE).

Tons.	A.	B.	C.	D.	E.	F.	G.	H.
$\frac{1}{2}$	$3\frac{13}{16}$	$2\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$	1	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{16}$
$\frac{3}{4}$	$4\frac{1}{8}$	$2\frac{1}{2}$	$1\frac{7}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{1}{4}$
1	$4\frac{1}{2}$	$2\frac{11}{16}$	$1\frac{5}{8}$	$1\frac{5}{16}$	$1\frac{1}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{5}{16}$
$1\frac{1}{2}$	$4\frac{7}{8}$	$2\frac{13}{16}$	$1\frac{3}{4}$	$1\frac{5}{16}$	$1\frac{3}{16}$	1	$\frac{7}{8}$	$\frac{3}{8}$
2	$5\frac{1}{4}$	3	2	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$\frac{15}{16}$	$\frac{7}{16}$
3	$5\frac{5}{8}$	$3\frac{1}{8}$	$2\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{5}{16}$	$1\frac{1}{8}$	1	$\frac{1}{2}$
4	$5\frac{15}{16}$	$3\frac{5}{16}$	$2\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{3}{8}$	$1\frac{3}{16}$	1	$\frac{9}{16}$
5	$6\frac{5}{16}$	$3\frac{7}{16}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{7}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$\frac{5}{8}$
6	$6\frac{21}{32}$	$3\frac{5}{8}$	$2\frac{11}{16}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$\frac{11}{16}$
8	$7\frac{3}{8}$	$3\frac{15}{16}$	3	$1\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$\frac{11}{16}$
10	$8\frac{1}{8}$	$4\frac{1}{4}$	$3\frac{5}{16}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{3}{8}$	$\frac{3}{4}$
12	$8\frac{27}{32}$	$4\frac{9}{16}$	$3\frac{5}{8}$	$1\frac{7}{8}$	$1\frac{7}{8}$	$1\frac{13}{16}$	$1\frac{7}{16}$	$\frac{3}{4}$
15	$9\frac{9}{16}$	$4\frac{7}{8}$	$3\frac{7}{8}$	$1\frac{15}{16}$	2	$1\frac{15}{16}$	$1\frac{9}{16}$	$\frac{13}{16}$
18	$10\frac{1}{4}$	$5\frac{3}{16}$	$4\frac{3}{16}$	$2\frac{1}{8}$	$2\frac{1}{8}$	$2\frac{1}{8}$	$1\frac{5}{8}$	$\frac{13}{16}$
21	11	$5\frac{1}{2}$	$4\frac{1}{2}$	$2\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{1}{4}$	$1\frac{3}{4}$	$\frac{7}{8}$

Taking O = 2, we have, by formula, *p.* 453, in tons and inches,

21 *tons hook:* $a = 9{\cdot}28$, $r = 5$, $Z = 6{\cdot}2$

$$\therefore\ f_t = \frac{21}{9{\cdot}28} + \frac{21 \times 5}{6{\cdot}2 \times 2} = 2{\cdot}26 + 8{\cdot}47 = \underline{10{\cdot}73 \text{ tons.}}$$

5 *tons hook:* $a = 3{\cdot}95$, $r = 2{\cdot}97$, $Z = 1{\cdot}32$

$$\therefore\ f_t = \frac{5}{3{\cdot}95} + \frac{5 \times 2{\cdot}97}{1{\cdot}32 \times 2} = 1{\cdot}27 + 5{\cdot}62 = \underline{6{\cdot}9 \text{ tons.}}$$

1 *ton hook:* $a = 2$, $r = 2{\cdot}15$, $Z = {\cdot}5$

$$\therefore\ f_t = \frac{1}{2} + \frac{1 \times 2{\cdot}15}{{\cdot}5 \times 2} = {\cdot}5 + 2{\cdot}15 = \underline{2{\cdot}65 \text{ tons.}}$$

Apparently, stresses of 10, 7, and 3 tons are experienced with the given loads, if the co-efficient O = 2 be used. But if the bending formula be true without O, stresses of $17\frac{1}{2}$, $12\frac{1}{2}$, and 5 tons are involved. Now the elastic limit for wrought iron is from 12 to 15 tons, so the bending theory would appear to be insufficient, and further that engineers are still wise in designing hooks and such constructions by reference to the breaking load, on which a factor of not less than 6 is adopted.

Example 31.—A longitudinal steel boiler stay, 20 ft. long and 2″ diameter, supports a flat area of 15 ins. sq., having on it a pressure of 120 lbs. per sq. in. Find the greatest stress in the stay due to its own weight and the steam pressure. (Hons. Mach. Constr. Exam. 1890.)

$$\text{Weight of stay } w = 20 \times 12 \times 3{\cdot}14 \times {\cdot}29 = 218{\cdot}5 \text{ lbs.}$$
$$\text{Steam pressure P} = 15 \times 15 \times 120 = 27{,}000 \text{ lbs.}$$
$$B_m = \frac{w\,l}{8} = f_o \frac{\pi d^3}{32} \quad \text{and } f_o = \frac{14\,w\,l}{11\,d^3} = 8342 \text{ lbs.}$$
$$f_t\,a = 27{,}000 \qquad \text{and } f_t = \frac{27{,}000}{3{\cdot}14} = 8598 \text{ lbs.}$$
$$\therefore \quad \underline{\text{Total stress} = f_t + f_o = 8598 + 8342 = 16{,}940 \text{ lbs.} = 7{\cdot}56 \text{ tons.}}$$

Example 32.—A piece of **T** iron consists of a web 4″ deep and ½″ thick, and a flange 2″ wide and ½″ thick. Compare its strength under longitudinal pull for the two cases (1) with line of action through centre of web depth; (2) with line of action passing through centre of figure of the **T**. (Hons. Mach. Constr. Ex. 1888.)

See Fig. 414.

Find neutral axis by taking moments round A : k = 1·75″. Draw lines of limiting stress and find stress areas.

$$Z = \text{area} \times \text{arm} = {\cdot}6875 \times 3{\cdot}205 = 2{\cdot}203$$
$$a = 3 \text{ sq. ins.} \quad \text{and } r = {\cdot}75''$$
$$\text{Case (1)} \quad f_{\max} = \frac{W}{3} + \frac{W \times {\cdot}75}{2{\cdot}203} = {\cdot}67\,W$$
$$\text{Case (2)} \quad f_{\max} = \frac{W}{3} = {\cdot}33\,W$$
$$\underline{\frac{\text{Strength (1)}}{\text{Strength (2)}} = \frac{{\cdot}33}{{\cdot}67} \quad \text{or as } 1 : 2 \text{ roughly}}$$

Combined Bending and Compression Stress-Action is calculated by the same formula as for tension and bending, by substituting f_c in the direct stress.

Example 33.—Fig. 415 shews a 'Fairbairn' crane. Draw the curve of bending moment for all sections, and design a suitable section at A B, taking $f_o = 5$ tons. (Hons. Mach. Constr. Ex. 1887.)

B_m diagram is given in Fig. 415, using centre line of jib as base line. At each section the

Moment = W × horizontal arm to axis of section.

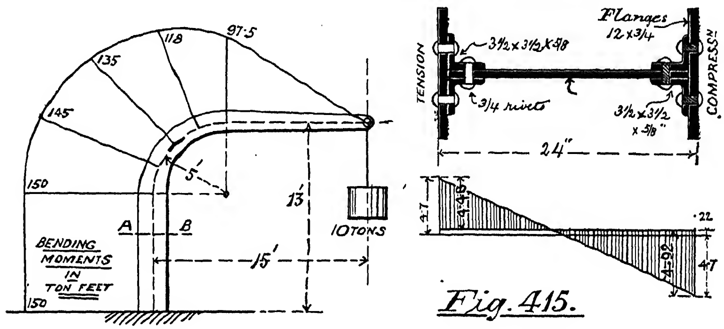


Fig. 415.

Section at A B can only be obtained by trial and error, and has thus been found in Fig. 415. Checking by approximate method :

$$\left.\begin{array}{c}\text{Area of two angles, one flange, and}\\ \text{portion of web between angles}\end{array}\right\} = 16 \text{ sq. ins.}$$

$$Z = a\,h = 16 \times 24 = 384 \quad \text{and total area} = 45 \text{ sq. ins.}$$

$$r = 15 \times 12 = 180''.$$

$$\therefore \quad f_{\max} = \frac{10}{45} + \frac{10 \times 180}{384 \times 1} = \cdot 22 + 4\cdot 7 = 4\cdot 92 \text{ tons.}$$

Ships' davits are similarly calculated, but their sections are like that of a crane hook, and the same precautions apply.

Strength of Pillars and Struts.—Although these fail by compression and bending, the action is not so simple. Struts having a length of ten or twelve times their diameter are reckoned for direct crushing only, but longer pillars bend before breaking. Euler* devised a formula to give the greatest load consistent with stability, that is, beyond which the bar could not restraighten. Let $Q = \frac{\pi^2 E I}{l^2}$. Then the stable loads w are given in the following table :—

* Pronounced 'Oyler.'

EULER'S FORMULÆ FOR LONG COLUMNS.

	One end fixed, the other free	$w = \frac{1}{4}\frac{\pi^2 E I}{l^2}$	$= \frac{1}{4} Q$
	Both ends free but load guided	$w = \frac{\pi^2 E I}{l^2}$	$= 1 Q$
	One end fixed the other free, but load guided	$w = 2\frac{\pi^2 E I}{l^2}$	$= 2 Q$
	Both ends fixed, and load guided	$w = 4\frac{\pi^2 E I}{l^2}$	$= 4 Q$

A factor of safety of 5 must be employed, and I can be found either from table (*p.* 429) or graphically (*p.* 431). The neutral axis for I must lie *across the greatest width of section.*

Euler's rules do not compare favourably with experiment, so engineers prefer Gordon's formulæ, which are a modified form of those made by Hodgkinson from his experiments. They give

the breaking stress only, and an arbitrary factor must be applied then.

(1) For pillars with both ends flat and carefully bedded

$$f \text{ breaking} = \frac{a}{1 + br^2}$$

(2) For pillars with both ends pointed or imperfectly fixed

$$f \text{ breaking} = \frac{a}{1 + 4br^2}$$

$r = \frac{l}{\text{shortest diameter}}$ and values of a and b are as follows:

Gordon's Constants.	Tons a	b
For solid or hollow round C. I. pillars	36	$\frac{1}{400}$
,, solid rectangular C. I. pillars	36	[illegible]
,, solid round W. I. pillars	16	[illegible]
,, solid rectangular W. I. pillars	16	[illegible]
,, pillars of **L T ⊓ + or H** section, W. I.	17	[illegible]
,, solid round pillars, mild steel	[illegible]	[illegible]
,, solid round pillars, strong steel	[illegible]	[illegible]
,, solid rectangular pillars, mild steel	30	[illegible]
,, solid rectangular pillars, strong steel	50	[illegible]

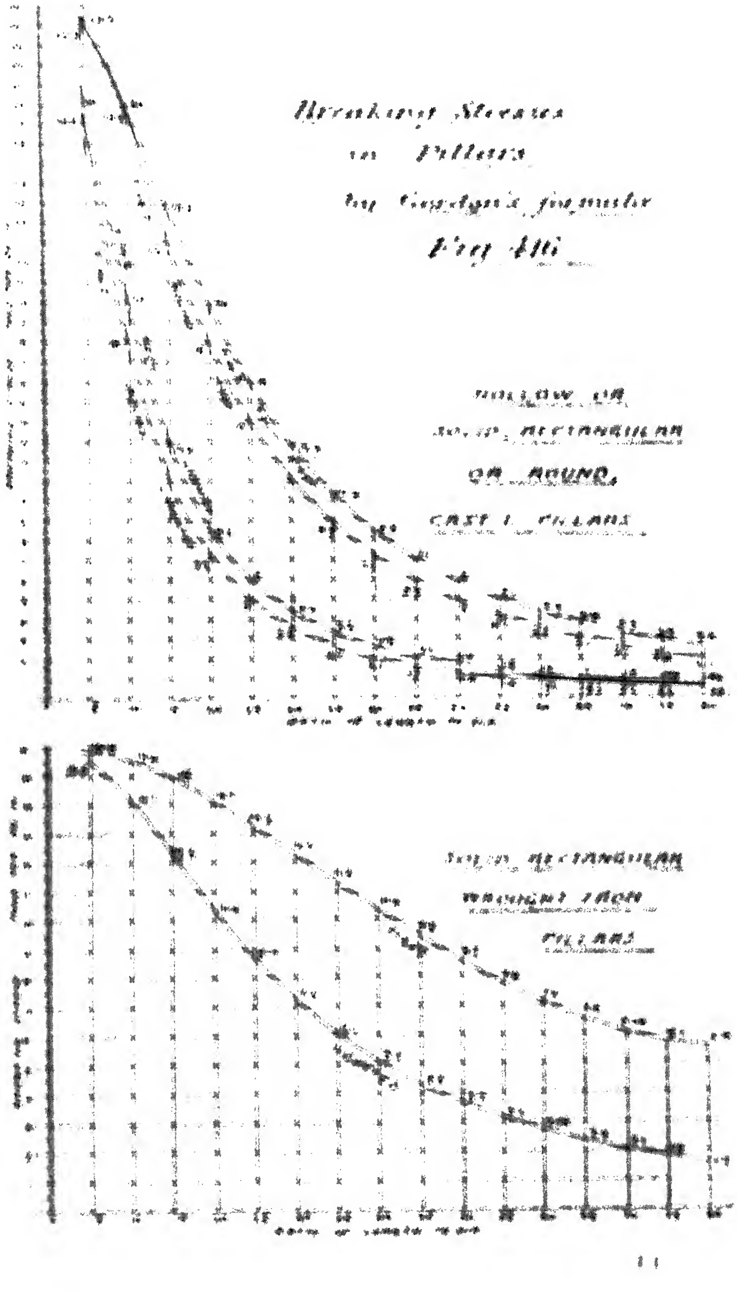
Some results from these formulæ are shewn in Fig. 416, and will be found handy for reference. A factor of safety of at least 6 must be used, and 10 or 12 in the case of moving parts. Then

$$W = \frac{f \text{ breaking} \times \text{area of section}}{\text{factor}} \text{ tons.}$$

Claxton Fidler demonstrates (Paper, Inst. C.E., 1886) that pillar strength "cannot be defined by any hard or fast line, but only by an area" (viz., that enclosed between his curve or Gordon's, and Euler's), "within which experimental results place themselves at haphazard." (*See Appendix II., p. 857.*)

Breaking Stresses in Pillars by Gordon's formula

Fig [illegible]



Example [illegible] the diameter of a steel [illegible] long for a maximum load of [illegible] tons [illegible]

[illegible]

$W = $ [illegible]

[illegible] and solving the quadratic

[illegible] at centre

For the small end $W = f_c a$

[illegible] and $d = $ [illegible] at small end

The [illegible] must be tapered as shown in Fig. 417 to meet the bending stress at the large end, due to pendulum motion.

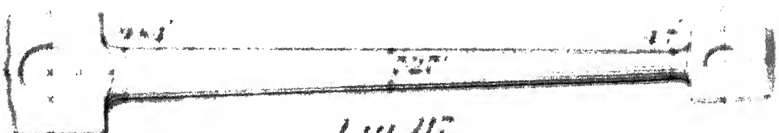


Fig. 417.

Strength of Furnace Tubes. No satisfactory theory having been propounded for these, we are driven to the adoption of empirical formulæ obtained experimentally. A tube pressed from outside is in a condition of unstable equilibrium, and if long enough fails by bulge or collapse. Then, for *plain unstrengthened tubes*, where length is from 400 t up to 10 d,

$$\text{Safe } p = \frac{1{,}600{,}000\, t^2}{l d} \Big\} \text{ By Fairbairn and Unwin.}$$

$$\text{Or safe } p = \frac{90{,}000\, t^2}{(l+1)\, d} \Big\} \text{ By Board of Trade.}$$

For tubes strengthened by noncollapse rings whose distance apart $= 400\,t$ or less, the stresses in the material are directly compressive, and bulging cannot occur. Then

$$\text{Safe } p = \frac{8000\, t}{d} \Big\} \text{ By Board of Trade.}$$

If the rings are [illegible] apart, the total [illegible] are [illegible], with [illegible] = distance between rings.

For [illegible] [illegible] rate $p = \frac{[illegible]}{d}$ By Board of Trade

where d = [illegible] diameter.

Combined Torsion and Bending [illegible] [illegible] [illegible] [illegible] [illegible]:

$$\text{In pure bending } f = \frac{B_m}{\frac{\pi d^3}{32}}$$

$$\text{and} \quad \text{In pure torsion } f_s = \frac{T_m}{\frac{\pi d^3}{16}} \qquad \textit{See also Appendix II., p. 8(9).}$$

Combining these into one *direct stress* f_s, it can be shown that

$f_s = \dfrac{B_m + \sqrt{B_m^2 + T_m^2}}{\frac{\pi d^3}{16}}$ which is the stress caused by a twisting moment $B_m + \sqrt{B_m^2 + T_m^2}$, or a bending moment of $\dfrac{B_m + \sqrt{B_m^2 + T_m^2}}{2}$.

In combined bending and torsion

(a) Equivalent twisting moment $= B_m + \sqrt{B_m^2 + T_m^2}$

(b) Equivalent bending moment $= \dfrac{B_m + \sqrt{B_m^2 + T_m^2}}{2}$

The first is most used. Let the shaft in Fig. 318 be under two pulls, P_t and W. B_m due to W is found by Case IV and set off at [illegible]. B_m due to P_t is shown by the diagram [illegible], where [illegible] is formed by the balancing force at [illegible], and action shown at [illegible]. [illegible] is the total B_m, having regard to sign. Twisting only occurs between W and [illegible], and is drawn at [illegible] to the same scale as the B_m. Now combine [illegible] to form the equivalent diagram *abcdefg*, every ordinate of which is obtained from the auxiliary diagram [illegible]. Thus, take B_m on [illegible] and T_m on [illegible], and, placing them at right angles, join the hypotenuse; then turn B_m round into line with the latter, and measure off the total ordinate upon [illegible]. The total shaded diagram thus obtained

is the *equivalent* T_m, and the darker diagram, half of this, is the *equivalent* B_m.

Example 35.—In Fig. 419 are some dimensions of a crank shaft. Let P = 1 ton when at right angles to plane ACB, T_m being balanced by a couple M at D. Find greatest $B_m + T_m$ and diameter of shaft when $f_s = 6$ tons. (Hons. Mach. Constr. Ex. 1893.)

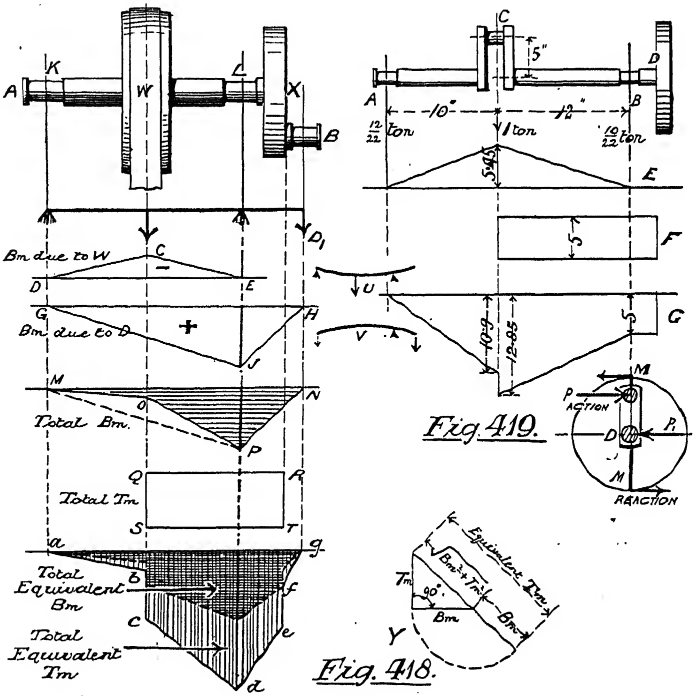


Fig. 419.

Fig. 418.

The end view shews how P_1 must be introduced to make the couple $P P_1$ complete. Then P_1 produces a B_m of $\frac{10}{22} \times 12 = 5{\cdot}45$ ton ins. as in diagram, and P gives a T_m of 5 ton inches.

$$\therefore \text{ Greatest equivalent } T_m = 5{\cdot}45 + \sqrt{54{\cdot}7} = 12{\cdot}85$$

and
$$d = \sqrt[3]{\frac{T_m\,16}{f_s\,\pi}} = \sqrt[3]{\frac{12{\cdot}85 \times 16 \times 7}{6 \times 22}} = 2{\cdot}2''$$

(*See p.* 997.)

Example [illegible] a force W [illegible] shaft equal to half the [illegible] Find [illegible] so that all shall be equally strong. If [illegible], find the [illegible]

[illegible]

Substituting [illegible]

[illegible]

Then [illegible]

Combined Torsion and Compression Stress Action.

When a shaft is under thrust and torsion at the same time, the stable load for the former is, by Euler

$$w = \frac{\pi^2 EI}{l^2}$$

and, if both actions be considered, according to Professor Greenhill

$$w_1 = \frac{\pi^2 EI}{l^2} - \frac{T_m^2}{4EI}$$

with which a factor of 5 must be adopted. Propeller shafts are examples.

Braced or Framed Structures. We commence the study of these by stating two rules.

Rule 1. If three oblique forces keep a body at rest, their directions meet at one point.

Rule 2. Their proportionate value will be shown by the respective sides of a triangle drawn parallel to the forces.

Let A and B (in Fig. [illegible]) represent two forces in direction and magnitude. Completing the parallelogram, R is the resultant,

[illegible]

[illegible]

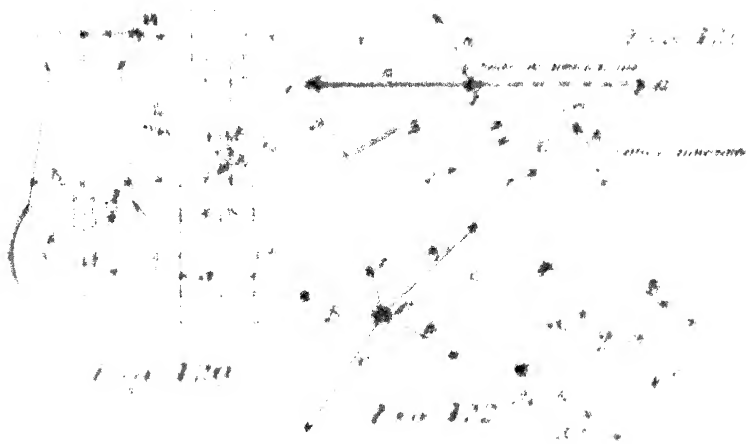


Fig. 130

Fig. 131

Fig. 132

Let forces [illegible] and [illegible] parallel to [illegible] respectively [illegible] the [illegible] [illegible] and [illegible]. Hence, *If a number of forces act at a point, their resultant magnitude will be shown by a polygon whose sides, taken in order, are drawn parallel to the forces.* If all the forces are known but two, these two can be found from the *polygon of forces.*

Maxwell's Reciprocal Stress Diagrams. The force diagrams of a framed structure [illegible] consisted of several polygons or [illegible], representing the stresses caused in the members by the action of the loads. It is known as the reciprocal stress diagram, and the extension is due to the late Professor Clerk Maxwell, though found independently by Mr W. P. Taylor. The method of lettering invented by Mr Bow is now the usual

[illegible] part of the system, which we shall now illustrate by a few examples.

Simple Roof Truss, Fig. 123. (1) *State all the external forces* (loads and reactions) in direction. The rafters divide their weights equally at a, b and a, c respectively. The reactions must balance the direct loads at b and c, and the load at a in addition, so that R_1 = [illegible] and R_2 = [illegible]. The ceiling weight does not affect the truss directly.

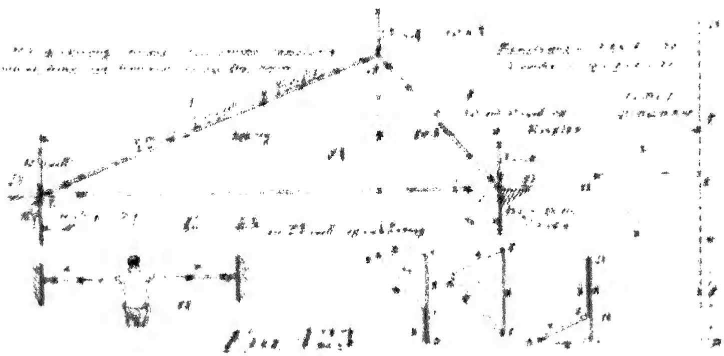


Fig. 123

(2) *Assign a letter to each cell.* Of these there is but one, the triangle A.

(3) *Place a letter in each division of the external space, as formed by the lines of forces.* These are at D, E, H, F, P in the figure.

(4) *Draw the force diagram for every set of radiating forces.* Take the four forces at corner c, each defined by the spacial letters thus: F D, D E, E A, A F. (Adhere to one method, preferably a right-handed rotation.) Set off the vertical E D in force diagram = [illegible] cwt. and D E = [illegible] cwt. Draw E A ‖ to bottom member bc. Then A F must be ‖ to ac, and must *meet at starting point* F. The steps are clearly shown in the figure at c. Notice that the arrows must *follow* round in the force diagram. The novice may imagine a dotted circle round each set of forces to avoid confusion caused by seeing two arrows on one member in opposite directions. The reason of the latter is explained at H, where the same pull is felt in opposite directions on the walls. If the

polygon does not properly close, the arrows may not be in the right direction, and a new examination must be made.

Next take the point *a*, and draw the triangle FA, AE, EF in the manner shewn at J. EF should measure 15 cwts. Finally, draw the polygon EA, AC, CD, DE for the point *b*, as shewn at K, making CD = 13·9 cwts. and DE = 10 cwts. And the stresses in the members may now be measured off, marking + for compression, and − for tension:

+ in AF = 14·5 cwts.
+ in AE = 10·4 cwts.
− in AC = 9·5 cwts.

Thick and thin lines also represent compression and tension respectively.

Suspension Bridge Chain.—A free uniform rope or chain hangs in a *catenary* curve, which is, however, so nearly like a parabola that the latter is always substituted for simplicity in practice. Taking the chain in Fig. 424, supposed weightless, but with loads at even distances as shewn, the forces at L and B are necessary to keep equilibrium, and the chains will be in tension as shewn by the arrows. Supposing reactions to be $3\frac{1}{2}$ each, the triangles ABC, ACD, AEF, &c., are drawn in succession. Then the distribution of load may be found for CD, DE, EF, &c., and the stresses in the chain also measured.

Warren Girder with Symmetrical Loads.—*First, Distributed on lower boom* (Fig. 425). The cells are equilateral triangles, and the girder has been much used for American bridges. Loads being 1, 1, 1, reactions are $1\frac{1}{2}+1\frac{1}{2}$, and the force $1\frac{1}{2}$ at J causes compressions in HB and HA, but tensions in AJ and AB. The force diagrams are drawn for points 1, 2, 3, 4, 5, &c., and the total diagram is given in the figures JKGDA. Measuring the latter, we find the stresses to be as follows:—

In AH and HG = 1·73 +	In HB and HF = 1·73 +
,, BA and GF = 1·73 −	,, HD = 2·31 +
,, CB and FE = 0·58 +	,, AJ and GK = 0·86 −
,, DC and ED = 0·58 −	,, CM and LE = 2·02 −

Next, let a Warren girder be loaded centrally on top boom. The diagram is found at Fig. 426, and can be easily followed.

Warren Girder with Unsymmetrical Load (Fig. 427)

The load is placed on the top boom and the force diagram shown

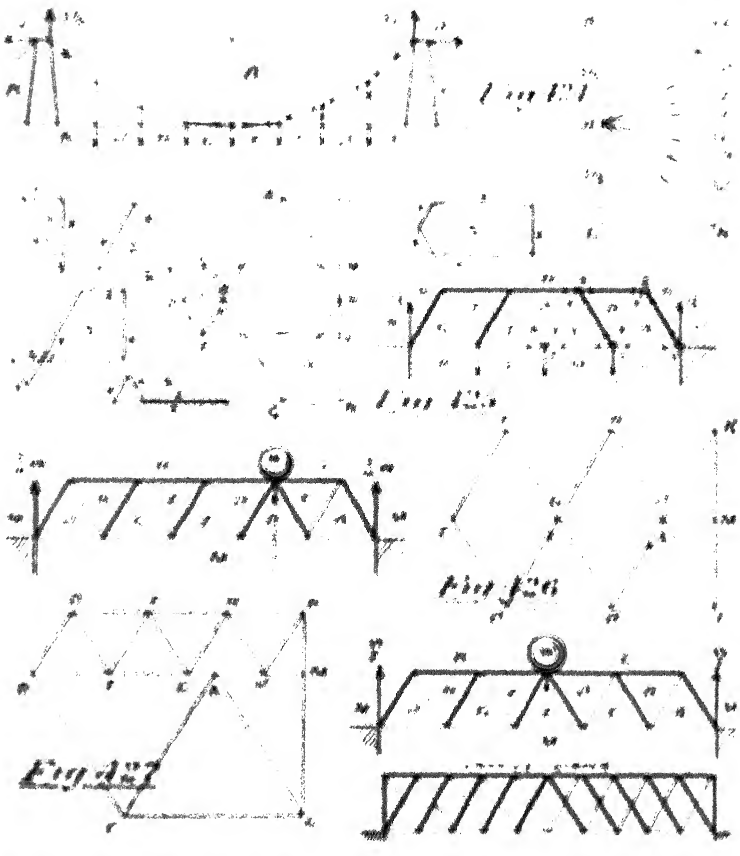


Fig. 424

Fig. 425

Fig. 426

Fig. 427

below, but may be worked for other conditions. R M is, of course, $\frac{7}{12}$ W, and M L $\frac{5}{12}$ W, and all the figures should close at R. If W be placed at each apex successively, and a separate diagram found for every case, we may, after examination, find the maximum stresses due to *rolling load*. Then Fig. 41?

gives stresses due to bridge-weight, and 427 those due to locomotive, &c. Tabulate the stresses so as to find the maxima, thus:

Member.	Rolling Load.					Maximum live load.		Stress due to dead load.	Total maximum.
	1st position	2nd position	3rd position	4th position	5th position	+	−		
AH	...	...	...	...	...	...	...	...	+
HG	...	...	...	...	...	...	...	...	+
BA	...	...	...	...	...	...	...	...	−
&c.									

After which the bars are designed to meet the stresses, either as ties or struts. A *lattice girder* is shewn in Fig. 426, being two Warren girders superposed.

Example 37.—The post, tie rod, and jib of a crane (Fig. 428) are 15, 45, and 50 feet long respectively. Find all the stresses, (1) with barrel on tie rod; (2) with barrel on jib, with a load of 5 tons. (Eng. Exam., 1887.)

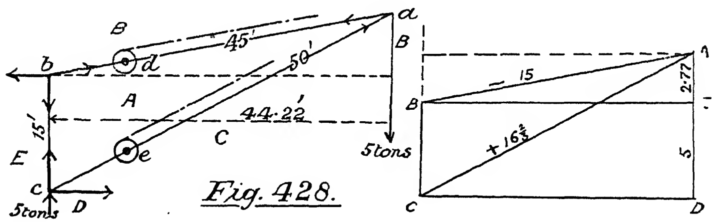


Fig. 428.

Reactions.—Load produces both turning effect and downward pull, resisted by equal horizontal forces at *b* and *c*, and by 5 tons upward force at *c*. That at *b* is supplied partly by bending strength of post, and partly by balance-weight. Letter the truss.

First, suppose the weight hung from *a* as a fixed point. Draw B C in stress diagram = 5 tons, and complete triangle B C A. Passing to *b*, E A must be a pull to produce equilibrium, and B A E is the force triangle. Finally, forces at *c* are shewn by polygon A C D E A, and stresses are:

on A B = 15 − on C A = 16·66+ and on A E = 2·77 −

Also E B = C D = 14·74.

and weight couple = righting couple

Tons. Feet.		Tons. Feet.
$5 \times 44{\cdot}22$	=	$14{\cdot}74 \times 15$
221·1	=	221·1

Case (1). Stress in tie rod from *d* to *a* becomes $-15 \ +5 = 10$ tons −
Case (2). Stress in jib from *e* to *a* becomes $+16\frac{2}{3}+5 = 21\frac{2}{3}$ tons +
other stresses being unaltered. The advantage of Case (1) is obvious. If the barrel be between *d* and *e* the stress must be resolved on the two members. As the load varies from nothing to a maximum, the righting moment of balance weight should be half the maximum.

Redundant Members are such as receive no stress according to force diagram, but contribute usually to resist buckling. In Fig. 429, cross-members connect weak strut B with strong tie A,

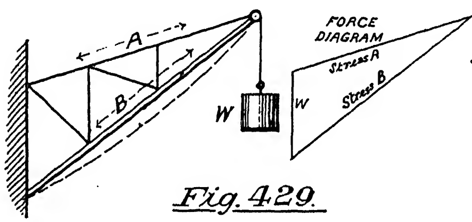


Fig. 429.

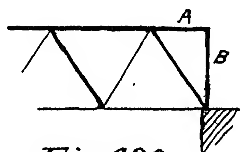


Fig. 430.

but otherwise receive no stress. In Fig. 430 A is redundant, but receives stress due to instability of strut B. (*See App. III., p.* 923.)

Example 38.—A crane is constructed as in Fig. 431. Draw the stress diagram for internal and external forces. (Hons. Mach. Constr. Ex. 1888.)

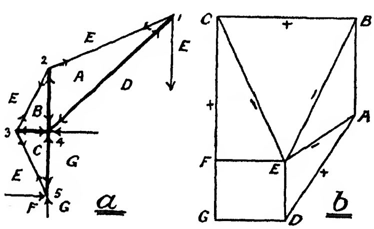


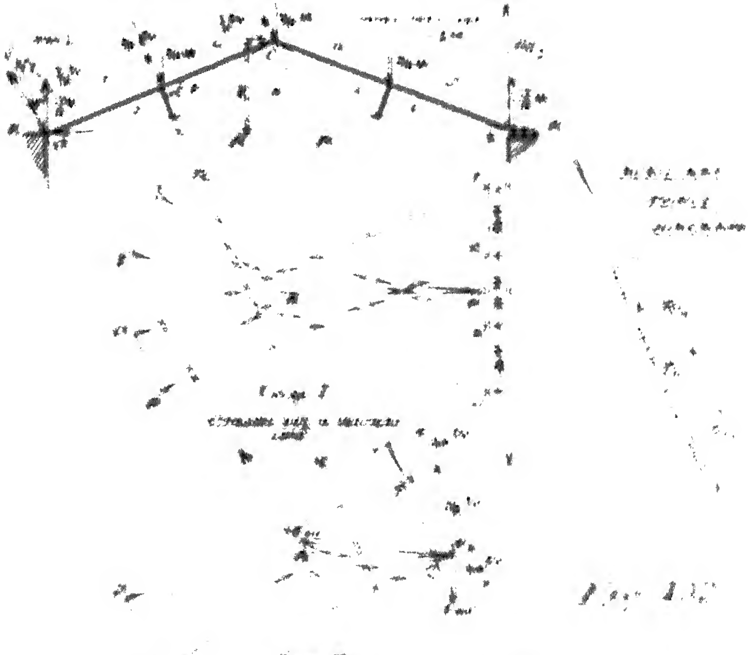
Fig. 431.

The crane is shewn at *a*, and the stress diagram found at *b*, commencing with the weight E D.

Roof truss with Five Cells [illegible]

[illegible]

Case I. [illegible]



Case II

Taking each side of roof [illegible] to be distributed as [illegible] W, [illegible] W, and [illegible] W at the three points respectively, making reactions = total load, [illegible] W, the forces are then as given, and after lettering the spaces, the stress diagram is found as below.

Case III. *Oblique load due to wind pressure.* Considering the wind to blow horizontally exerting a force of [illegible]

60 lbs. per sq. ft. (according to the exposure) upon the area ($k \times$ width of bay).

$$\text{Total force } P_t = 56 \times k\,w', \quad \text{say, in lbs.}$$

Then P_t may be resolved into two forces, one parallel to the rafter, and one (P_n) normally, and

$$P_n \text{ total} = \frac{P_t \times k}{a\,c}$$

This force is distributed at a, d, and c as $\frac{3}{16}\,P_n$, $\frac{5}{8}\,P_n$, $\frac{3}{16}\,P_n$. In iron roofs the expansion is usually allowed for by fixing one end (a) and leaving the other free (b), which allows us to say the reaction, R_{t_2}, is vertical. Now R_{t_1}, R_{t_2}, P_n are three balancing forces, and must meet in one point X, found by producing P_n to meet R_{t_2} produced. Then joining a X, the direction of R_{t_1} is found, and the amounts R_{t_1} and R_{t_2} further obtained from the auxiliary diagram. If the wind blow from the right, P_n is exerted on $c\,b$, and x will be above instead of below b (Case III.). Both II. and III. must be examined, though we only have space for Case II. Take the lettering as in I., with the exception of the additional external space L, and draw the stress diagrams as shewn below.

Finally, tabulate the stresses for Cases I., II., and III.: then add I. to II. and III. separately to find the maximum stresses, and design the members to suit. (*See Appendix IV., p.* 953.)

Framed Structures of Three Dimensions are such as include a solid instead of an area. They must be solved by a step-by-step process, taking each plane in succession. We will explain by means of an example.

Example 39.—A sheer legs (Fig. 433) is formed of two fore legs 145′ long and 60′ apart at the base, and a back leg 170′ long attached to a nut having a travel of 40′. The maximum overhang is 40′, and load 100 tons. Find the stresses in the members, (1) and (2), at each end of the nut stroke; (3) when the load is directly over the base plates. (Hons. Mach. Constr. Ex. 1888.)

Case I. Nut at D.—Taking first the plane A D B, the diagram P shews stresses in A B = 174 tons, in A D = 82 tons.

Turning to the end view, the stress of 174 in A B must be resolved in each leg, as in diagram Q, giving stresses

in $a\,b$ and $a\,c$, each = 88 tons.

Case II. Nut at E.—R is the first diagram and S the second giving stresses

in $A_1B = 92$ tons, and in ab and ac each $= 47$ tons.

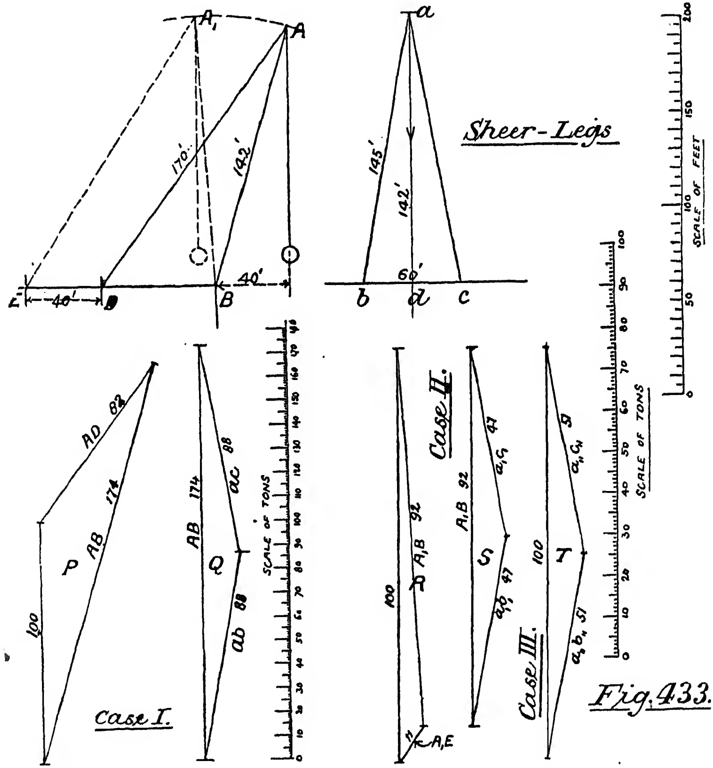


Fig. 433.

Case III. is worked entirely from the end view. 100 tons is to be distributed on the fore legs, causing no stress in the back leg. Then by diagram T, stresses are

in ab and ac each $-$ 51 tons.

CHAPTER IX.

ON ENERGY, AND THE TRANSMISSION OF POWER TO MACHINES.

WE commence with a few definitions and explanations.

Force and Mass.—Engineers use 'gravity' units for these: the unit of force being 1 lb. and that of mass 32·2 lbs. (g) or:

$$\text{mass} = \frac{w}{g}$$

Velocity is estimated in feet per second. If *uniform*, the distance travelled (s) depends both on rate and time occupied: thus,

$$s = tv \quad \text{(distance = time × velocity)}$$

shewn graphically at A, Fig. 434. The area B shews similarly the distance travelled, under *variable velocity* given by the curve; the areas being measured as at Fig. 326, Chapter VIII.

Acceleration (f) is the increase of velocity during each second. *Uniform acceleration* is produced by any constant force, the latter being measured by the increase of momentum it produces.* Momentum = mass × velocity.

$$\therefore \text{Force producing acceleration} = \frac{w}{g} \times f.$$

Uniformly Accelerated Velocity.—A body starting from rest at O (C, Fig. 434) has its velocity gradually increased by the amount f during each second t, and the final velocity is $4f$. But the total time is 4. Therefore final velocity,

$$v = ft \ldots\ldots (1)$$

and the distance is shewn by the area under the velocity curve at C, or:

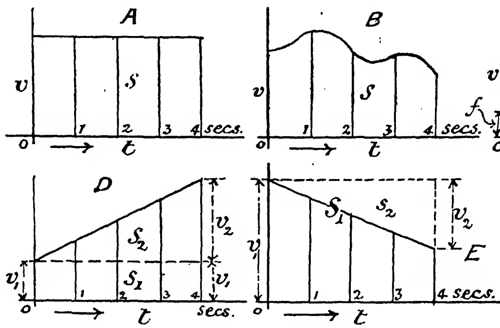
$$s = \frac{v}{2}t = \tfrac{1}{2}ft^2 \ldots\ldots (2)$$

Substituting value of t from (1) we have:

$$s = \frac{v^2}{2f} \text{ and } v^2 = 2fs \ldots\ldots (3)$$

* Newton's second Law.

With an original velocity v the distance is found by adding the two areas at D, Fig 434. (*See p.* 1099.)



Velocity curves.

Fig. 434.

Uniformly Retarded Velocity is a similar case to D, the final velocity and total distance being found by *subtraction* of areas, as at E, and are $v_1 - v_2$ and $s_1 - s_2$ respectively.

Collating results, with v_1 as original and v_2 as final velocities,

U.A.V. from rest.	U.A.V. with original velocity.	U.R.V.
$v_2 = ft$	$v_2 = v_1 + ft$	$v_2 = v_1 - ft$
$s = \frac{1}{2}ft^2$	$s = tv_1 + \frac{1}{2}ft^2$	$s = tv_1 - \frac{1}{2}ft^2$
$v_2^2 = 2fs$	$v_2^2 = v_1^2 + 2fs$	$v_2^2 = v_1^2 - 2fs$

Example 40.—A locomotive and train weighing 100 tons start on a level, and attain a speed of 60 miles per hour within one minute. What was the mean pull exerted?

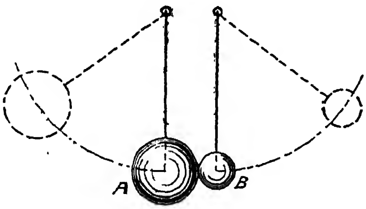
$$\text{From (1)} \quad f = \frac{v}{t} = \frac{60 \times 5280}{1 \times 60 \times 3600} = \frac{22}{15} \qquad (\textit{See p. } 1096.)$$

$$\text{and Pull in lbs.} = \frac{wf}{g} = \frac{100 \times 2240 \times 22}{32 \times 15} = 10266 \text{ lbs.}$$

or 4·6 tons, neglecting friction.

Conservation of Momentum.—Two balls, A and B, Fig. 435, raised simultaneously, are allowed to fall, strike, and rebound. The duration of shock is called the impact, and it is

found that the added momenta of the balls is the same whether before or after impact, a fact useful in many calculations. In the case of ordnance the total momentum is divided at explosion,



Momentum

Fig. 435.

equally between gun and carriage on the one hand, and the shot and charge on the other. Introducing a practical coefficient 1·1,

$$\left.\begin{array}{c}\text{Highest velocity}\\ \text{of recoil}\end{array}\right\} = \frac{(\text{weight of shot and charge}) \times (\text{muzzle velocity}) \times 1{\cdot}1}{(\text{weight of gun and carriage})}$$

the quantities being in lbs. feet and seconds.

But $p = \dfrac{w\,v}{gt}$ (pp. 473–4) $\therefore$ mean force of recoil $= \dfrac{w\,v}{gt}$ and maximum force = mean force × 2.

Energy is the capacity to do work.—*Potential Energy* is latent till some small change occurs to give it actual value: thus the chemical energy in coal requires a small starting heat, and the water in a high tank may be released by opening a small valve. *Kinetic Energy* or energy due to motion, is always visible so to speak, except in the case of molecular movement merely.

EXAMPLES OF ENERGY FORMS.

Potential Energy.	1. Raised weight (solid or liquid):	Energy of position.
	2. Clock spring wound up: bent bow: 3. Compressed gas:	Elastic Energy.
	4. Nerve Energy:	(Capable of muscular exertion.)
	5. Electrical Energy:	That due to separation of positively and negatively-electrified bodies, as in frictional electricity.
	6. Chemical Energy:	Due to separate existence of elements, as in gunpowder and coal.

EXAMPLES OF ENERGY FORMS (*continued*).

Kinetic Energy.	7. Muscular Energy :	(When in motion.)
	8. Gas Expansion :	*e.g.*, the wind, heat engines.
	9. Mechanical Energy :	As in machines.
	10. Electrical Energy :	The current in motion, as in Voltaic and Faradaic electricity.
	11. Heat Energy :	Being molecular motion.
	12. Chemical Energy :	When combining, on account of affinity of elements.
	13. Radiant Energy :	The vibration of the ether causing light and heat.

The true energy is that only which is *available*, by reason of a certain difference of 'pressure,' 'head,' or 'potential,' as measured within fullest attainable limits.

NATURE'S STORES OF ENERGY.

I. Heat Energy :	Direct from sun : probably sustained by meteoric impact.
II. Water Energy :	Due to fall from mountains to sea.
III. Wind Energy :	Due to difference of pressure caused by sun's heat.
IV. Coal Energy :	Due to chemical condition of separation.
V. Petroleum or oil Energy :	Ditto.
VI. Tidal Energy :	Due to moon's attraction, principally.
VII. Electrical Energy :	(1.) Due to separation of kind, as in thunder clouds, and untractable : (2.) Due to very small differences of potential in both air and earth, and valueless for large operations.
VIII. Food Energy :	Due to sun's action on growth of plants.

All these, excepting VI., are due to the sun's heat, which has grown coal forests and daily evaporates water. V. is probably due to a condensation of the once glowing earth.

Conservation of Energy. In every system, the total energy, however changed in form, remains constant. This is shown by every-day experience, and although impossible to prove directly, is [illegible]. Stated generally,

$$\text{Kinetic Energy} + \text{Potential Energy} = \text{Constant},$$

but as its more useful form for the engineer in any machine,

$$\text{Total energy deposited} = \frac{\text{useful work}}{\text{given out}} + \frac{\text{work lost in}}{\text{resistances}},$$

unless some portion is stored for future use, as by a spring. The most usual form of loss is frictional heat, and we are quite certain that never more can be received than was first deposited, which at once disposes the claims of perpetual motion machines, depending as they do upon a surplus.

Transformation of Energy. Thus, generally, potential energy becomes kinetic, and *vice versa*, the simplest example being a pendulum which is alternately stationary but raised, and moving but fallen. Coal energy, potential in the mine, becomes kinetic, as heat, in the boiler, and kinetic, as mechanical energy, in the engine. Chemical energy becomes electric in the galvanic battery, and heat energy electric in the thermopile, while water may turn a dynamo through a turbine. A locomotive brake block converts mechanical energy into heat, and many other examples will suggest themselves.

Numerical Estimate of Various Energies. A raised weight may do work in falling. Therefore its

$$\underline{\text{energy in foot pounds} = wH \text{ (potential)}.}$$

When reaching the ground its velocity will be (*see bottom of p. 474*)

$$v = \sqrt{2gH} \quad \text{and} \quad H = \frac{v^2}{2g}.$$

Substituting this value in the first formula we have

$$\underline{\text{energy in foot pounds} = \frac{wv^2}{2g} \text{ (kinetic)},}$$

which may be applied to all cases of moving bodies, whatever the cause of their motion, for we may always *suppose* that the velocity has been caused by gravity, a strictly tenable artifice.

[illegible]

[illegible]

where [illegible] is the weight [illegible] R = average radius. Here energy [illegible]

Example [illegible]

[illegible] kinetic energy [illegible] the energy [illegible] greatest and [illegible] when the fluctuation [illegible] energy [illegible] the energy of [illegible]. Here Work [illegible] 388:

Energy per rev. [illegible] foot pounds.

K.E. energy of fly wheel [illegible] foot pounds

Ratio [illegible] fly wheel [illegible]

Let [illegible] highest vel. [illegible] least vel. [illegible]

Then [illegible]

Fluctuation [illegible] and [illegible]

[illegible]

But [illegible]

and [illegible]

Finally [illegible]

Highest revs. per m. [illegible] and least revs. [illegible]

The energy of a spring or compressed gas is the average force multiplied by the distance moved. Thus

Energy of compressed spring in foot pounds = $\frac{\text{greatest load in lbs.}}{2}$ × compression in feet

Energy of compressed gas in foot pounds = mean effective of total pressure [illegible] × expansion or stroke of piston in feet

A *unit of heat* will raise 1 lb. of water through 1° Fahr. when near 39°, its greatest density; and Dr. Joule found by experiment that

One unit of heat = 772·55 foot pounds of mechanical energy.

The number 772 therefore is spoken of as Joule's equivalent (J). (*See Appendix III., p.* 930.)

Electrical energy may be estimated in terms of mechanical energy as follows:

Energy in foot pounds = ·737 E Q.

where E = Electro-motive force in volts.

and Q = Quantity in coulombs. (*See App. III., p.* 926).

Lastly, Chemical energy is measured by its heating effect, found by careful experiment. Thus, 1 lb. of good dry bituminous coal will give out 14,500 units of heat when completely burnt, which may be further represented in foot pounds.

Prime Movers are machines which obtain Nature's energy at first hand for transmission of, or transformation into mechanical energy. Such are: Heat Engines, Water-Wheels and Turbines, Windmills, Electric Engines,* and Tidal Motors.

Power is direct or controlled energy, as distinguished from the free energy of Nature or that, say, of a bullet. The term has been more usually applied to mechanical energy or the mechanical equivalent of other energies, but is gradually being restricted to indicate 'rate of doing work,' one horse-power being the unit.

✓ **Transmitters of Power** remove the mechanical energy of a prime mover to a distance, or change the components and perhaps the whole form of the energy. The following is a list:—

1. Linkwork:	Connecting rods, coupling rods, cams and levers.
2. Shafting:	Lines of shafting, with clutches, couplings, and bearings.
3. Spur gearing:	For connecting parallel shafts.
4. Bevel gearing:	Connecting shafts at various angles.
5. Worm gearing:	Connecting shafts at right angles.
6. Belt gearing:	Connecting shafts at various angles. but chiefly parallel.
7. Rope gearing (cotton):	For high speeds.

* By voltaic battery or thermopile.

8 [illegible] — [illegible]
9 [illegible] — [illegible]
10 [illegible] — [illegible]
11 Compressed air — [illegible]
12 Hydraulic [illegible] — Water power [illegible]
13 [illegible] — [illegible]

Comparison of Agents. [illegible] transmitted [illegible] standard *Horse power* [illegible] *minute.* [illegible]

Horse power = [illegible]

In this way the following [illegible] have been calculated:

Horse Power of Various Agents

	[illegible]	[illegible]
A man raising his own weight vertically during a day of eight hours	[illegible]	[illegible]
[illegible] pushing and pulling at capstan	[illegible]	[illegible]
[illegible] turning a winch	[illegible]	[illegible]
Horse pulling a cart	[illegible]	[illegible]

[illegible] a man performs [illegible] and a good horse [illegible] of a horse power.

Theory of Machines. *A machine is an assemblage of parts whose relative motions are fully constrained, and whose purpose is the transmission or modification of energy.* The [illegible] method has been to refer all machines, however complicated, to six simple cases,* each consisting, according to [illegible], of more than one part. They are:

1. The Lever — [illegible]
2. The Wheel and axle — [illegible]
3. The Pulley — [illegible]
4. The Inclined Plane — sliding piece and [illegible]
5. The Wedge — [illegible] equivalent to double inclined plane
6. The Screw — screw and nut, equivalent to [illegible] inclined plane

* Called (wrongly) "Mechanical Powers."

and they can all be placed under two divisions—levers and inclined planes. There is always a point P where the energy enters, and a point W where it is removed,* and the *Principle of Work* states that

$$\text{Work put in at P} = \text{work taken out at W}$$

neglecting resistances. But as work = force × distance,

$$P \times d = W \times D, \qquad \text{or} \qquad \frac{W}{P} : \frac{d}{D},$$

where d = distance travelled by P, and D that travelled by W.

This is the underlying principle, and our investigations on machines are for the purpose of finding the comparison of the *distances or speeds* at P and W, for by inversion we shall obtain the relation of the *forces* W and P. The first is the ratio of *virtual velocities* and the second is *mechanical advantage*. Then, generally,

$$\frac{\text{vel. P}}{\text{vel. W}} = \frac{\text{force W}}{\text{force P}} = \text{Mech. Adv. } \frac{W}{P}*$$

The Lever is shewn under various forms in Fig. 436. By moments:

$$Pa = WA \qquad \text{and} \qquad \text{Mech. Adv.} = \frac{W}{P} = \frac{a}{A}$$ (*See Appendix I., p.* 763.)

The Wheel and Axle, Fig. 437, is reckoned similarly, and its

$$\text{Mech. Adv.} = \frac{a}{A} = \frac{\text{handle}}{\text{barrel rad.}}$$

A *train of gearing* in Fig. 438 consists of two pairs of wheels, a handle, and a barrel. The advantage of the first pair would be $\frac{a}{A}$: of the second pair $\frac{a_1}{A_1}$: and of the wheel and axle $\frac{a_2}{A_2}$. So the total

$$\text{Mech. Adv. } \frac{W}{P} = \frac{a}{A} \times \frac{a_1}{A_1} \times \frac{a_2}{A_2}$$

* The old letters P and W being retained, are meant to represent the *forces* and also the points of application. Rankine called them *effort* and *resistance* respectively. Note that frictional and other losses are entirely neglected on pp. 481–4 and *Theoretical* mechanical advantage is therefore the result.

(*See p.* 954.)

which can be easily proved by the levers shewn below. Generally, then, for toothed gearing with wheel and axle,

$$\text{Mech. Adv. } \frac{W}{P} = \frac{\text{followers}}{\text{drivers}} \times \frac{\text{handle rad.}}{\text{barrel rad.}}$$

the wheels being estimated by teeth, radius, or diameter.

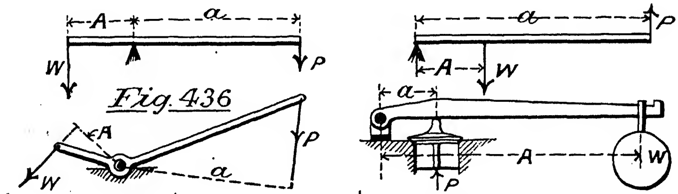


Fig. 436

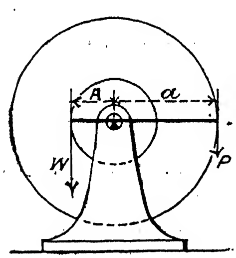


Fig. 437

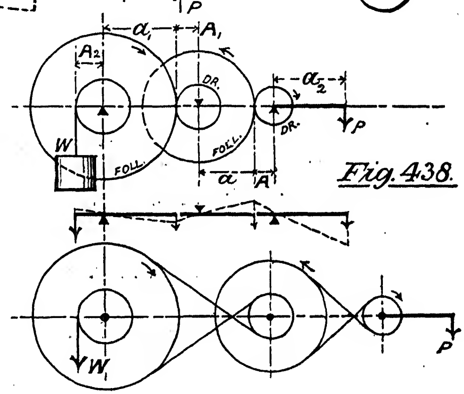


Fig. 438

Belting is a substitute for toothed gearing, as shewn in the lower diagram, a crossed belt giving the same direction of motion as one pair of wheels. N.B.—If speeds only are to be reckoned, the wheel and axle does not enter into the calculation.

The Block and Tackle (Fig. 439).—Neglecting friction, the cord has the same tension throughout, and there are (in the case shewn) six pulls on the weight, each equal to P.

$$\therefore \text{ Mech. Adv. } \frac{W}{P} = \frac{6}{1} \text{ or generally } = \frac{\text{No. of cords}}{1}$$

In any movable pulley M, $\frac{W}{P} = \frac{2}{1}$ because W only rises half the height of P, as shewn. (*See p.* 741.)

[illegible] *Compound Wheel and Axle* is given in its most useful [illegible] at Fig 446, p 314. P is the hand and W hangs from the [illegible]. Imagining the upper pulley to make one revolution

$$\text{Mech Adv} = \frac{W}{P} = \frac{P\text{'s dist}}{W\text{'s dist}} = [illegible] = \frac{2A}{A - B}$$

The reason W rises only half the difference of circumference is that the lower pulley is movable.

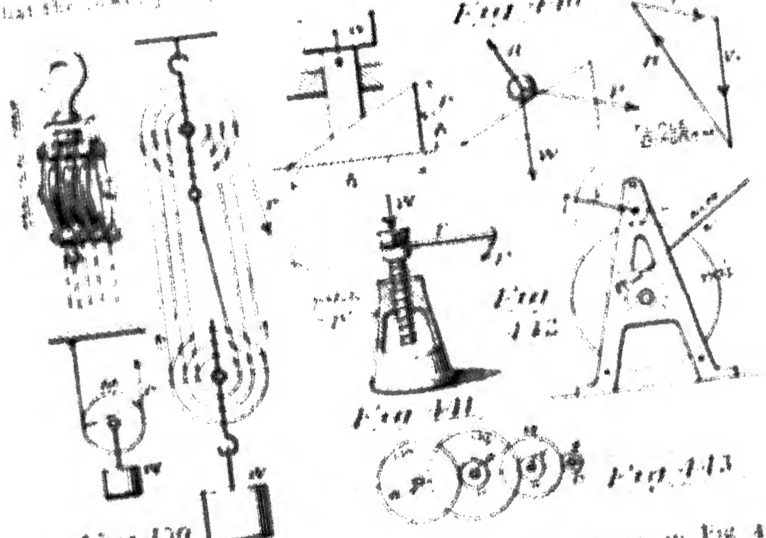


Fig 440

Fig 442

Fig 441

Fig 443

Fig 439

The Inclined Plane and Wedge are shown in Fig 440. While P moves through l, W is lifted through h, and

$$\text{Mech Adv} \; \frac{W}{P} = \frac{l}{h}$$

Or, a body being held on the plane by the three forces P, W, & R (the latter being normal), the relation of the three may be found by the triangle of forces, Fig 421. (See also p 464.)

The Screw exists in combination with the lever, as in Fig 441 (see also pp 206 71). If P make one rotation,

$$\text{Mech Adv} \; \frac{W}{P} = \frac{P\text{'s dist}}{W\text{'s dist}} = \frac{2\pi r}{p}$$

Example 42.—Arrange the gearing of a single purchase crab so that 60 lbs. on a 15″ handle may raise half a ton from a barrel 10″ dia., *neglecting friction.* (Eng. Ex., 1891.)

$$\text{Mech. adv. } \frac{W}{P} = \frac{\text{follower}}{\text{driver}} \times \frac{\text{handle}}{\text{barrel rad.}} = \frac{1120}{60}$$

$$\therefore \frac{1120}{60} = \frac{\text{follower}}{\text{driver}} \times \frac{15}{5} \qquad \therefore \frac{\text{follower}}{\text{driver}} = \frac{6\cdot22}{1}$$

So the pitch line diameters may be 6″ and 37·32″ for pinion and wheel respectively, as in Fig. 442. (*See p.* 955.)

Example 43.—A shaft A has a spur wheel of 120 teeth, which drives a pinion B with 11 teeth. On shaft B is a wheel of 132 teeth driving a pinion C of 10 teeth. Lastly, on shaft C is a wheel of 48 teeth driving a pinion D of 8 teeth. A turns at 2 revs. per m. Find speed of D. (Eng. Ex., 1885.)

The wheels are shewn in Fig. 443.

$$\frac{\text{vel. D}}{\text{vel. A}} = \frac{\text{followers}}{\text{drivers}} = \frac{120 \times 132 \times 48}{11 \times 10 \times 8} = \frac{864}{1}$$

$$\therefore \text{ D makes } 864 \times 2 = \underline{1728 \text{ revs. per m.}}$$

Example 44.—Two men at a crab exert 60 lbs. each on a 16″ handle. The pinion has 12 teeth, the wheel 72 teeth, and the chain barrel is 12″ diameter. Find the load raised, *neglecting friction.* (Eng. Ex., 1888.)

$$\text{Mech. adv. } \frac{W}{P} = \frac{\text{followers} \times \text{handle}}{\text{drivers} \times \text{barrel rod.}} = \frac{72 \times 16}{12 \times 6} = \frac{16}{1}$$

$$\therefore \underline{W} = 16 \times P = 16 \times 120 = \underline{1920 \text{ lbs.}}$$ (*See p.* 955.)

Example 45.—In a Weston block the diameter of the large sheave is 10″, and that of the smaller 9″. Find the load raised by a pull of 50 lbs., *neglecting friction.*

$$\text{Mech. adv. } \frac{W}{P} = \frac{2\,A}{A - B} = \frac{2 \times 10}{1} = \frac{20}{1}$$

$$\therefore \underline{W = P \times 20 = 1000 \text{ lbs.}}$$

(*See Appendix IV., p.* 954, *without fail.*)

Change Wheels in Screw-cutting.—General principles are explained at pp. 147 and 212, it being shewn that:

$$\frac{\text{Revolutions of mandrel}}{\text{Revolutions of leading screw}} = \frac{\text{No. of threads per inch. on mandrel}}{\text{No. of threads per in. on leading screw}}$$

in order to cut a definite pitch. This may also be stated as

$$\frac{\text{followers at L. S. end}}{\text{drivers at M end}} = \frac{\text{pitch L. S.}}{\text{pitch M screw}}$$

or the *pitches and wheels are in the same ratio*, which ratio, being found, must be accommodated by a suitable train.

Example 46.—In Plate V., the leading screw being $\frac{3}{4}''$ pitch, and the wheels in the set rising by 5 at a time from 20 to 120 teeth, it is required to arrange wheels to cut (1) a screw of 10 threads per inch, right-handed, and (2) a screw of $1''$ pitch left-handed.

(1) $\underline{\text{pitch ratio}}\ \frac{\text{L.S.}}{\text{M}} = \frac{\frac{3}{4}}{\frac{1}{10}} = \frac{15}{2}$

Putting 30 teeth on n (Fig. 135) into 75 on stud (b, Fig. 140): 30 teeth on stud into 90 on L.S., we have,

$\underline{\text{wheel ratio}} = \frac{75 \times 90}{30 \times 30} = \frac{15}{2}$ and the handle at n must be *down.*

(2) $\underline{\text{pitch ratio}}\ \frac{\text{L.S.}}{\text{M}} = \frac{\frac{3}{4}}{1} = \frac{3}{4}$

Putting 45 teeth on L.S. and 60 teeth on n; with any intermediate on stud (say 60) we have,

$\underline{\text{wheel ratio}} = \frac{45}{60} = \frac{3}{4}$ and the handle at n must be *up.*

Kinematics (of Machines) is a method of attacking machine problems devised by Prof. Reuleaux, and anglicised by Prof. Kennedy. We shall proceed to discuss its principles.

Pairs.—The constraining parts are termed pairs because they always occur in sets of two. Of these there are *higher* and *lower pairs.* The former connect by points or lines, but the latter by their whole surfaces.

Three kinds of lower pairs are possible: I. Sliding pairs, as a piston and cylinder. II. Turning pairs, as a journal or pin. III. Screw pairs, including all screws and nuts. Complete or *closed* pairs have their motions fully defined: *incomplete* pairs require further closure, as at Fig. 444, where gravity is not for the moment considered.

Kinematic Chains [illegible] connected [illegible]

The Slider-crank Chain [illegible] these the [illegible]

Fig. 111

Fig. [illegible]

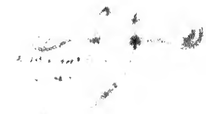


Fig. [illegible]

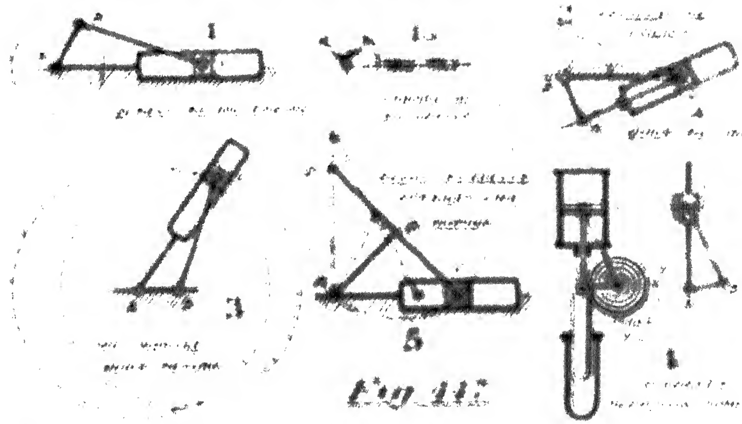


Fig. [illegible]

useful contrivances as in Fig. [illegible] [illegible] by inversion or change of the fixed link, and [illegible] the relative lengths. Thus

1. Fixing a [illegible] gives Direct acting engine.
2. Fixing b [illegible] Oscillating engine and Quick return [illegible] Plate X.)
3. Fixing a [illegible] Whitworth's Quick return (Fig. [illegible] Plate XI.)
4. Fixing block [illegible] [illegible] pendulum pump.
5. Fixing [illegible] [illegible] Russell's straight line motion, [illegible] making a straight line

[illegible] have their relative motions fully constrained by [illegible]. The first occurs at 4, 6, and 2, Fig. [illegible], [illegible] at 3 and 5, and [illegible] at 2. But at 4 and 2, [illegible], and at 3, dead points occur which must be overcome by the wheel or other force closure, unless an arrangement like [illegible] be employed, which shews coupled cranks at right angles [illegible]. [illegible] is often the closing force, *e.g.*, planing machine table, and many [illegible].

The Double Slider Crank Chain has three links, two turning pairs and two sliding pairs variously connected. Taking the [illegible] forms in Fig. [illegible],

1. Fixing [illegible]	gives	Donkey pump mechanism.
2. Fixing [illegible] and [illegible] at right angles, and [illegible] turning pair [illegible]	,,	Elliptic trammels, oval chuck, and Oldham's coupling.
3. Fixing [illegible] and [illegible] at right angles, [illegible] one turning pair at [illegible], two sliding and one turning pair at [illegible]	,,	Rapson's slide (giving an increased leverage as the tiller is moved hard over).

The Quadric Crank Chain, Fig. 419, has four links and four turning pairs. (*See Appendix I., p. 764.*)

1. Fixing [illegible]	gives	Beam engine; force closure by fly wheel.
2. Fixing [illegible], and making [illegible]	,,	Watt's parallel motion.
3. Fixing [illegible] but making opposite links equal	,,	Wheel coupling gear for locomotive closure by double chain.
4. Ditto, but altering lengths	,,	Special motion in wire rope making (preserving verticality of drums).
5. Ditto, but doubling the chain as shewn	,,	Roberval's balance (allowing weight to be placed anywhere on pan).

Most lower-paired chains can be reduced to these three cases, which shews the advantage of discussing mechanism kinematically.

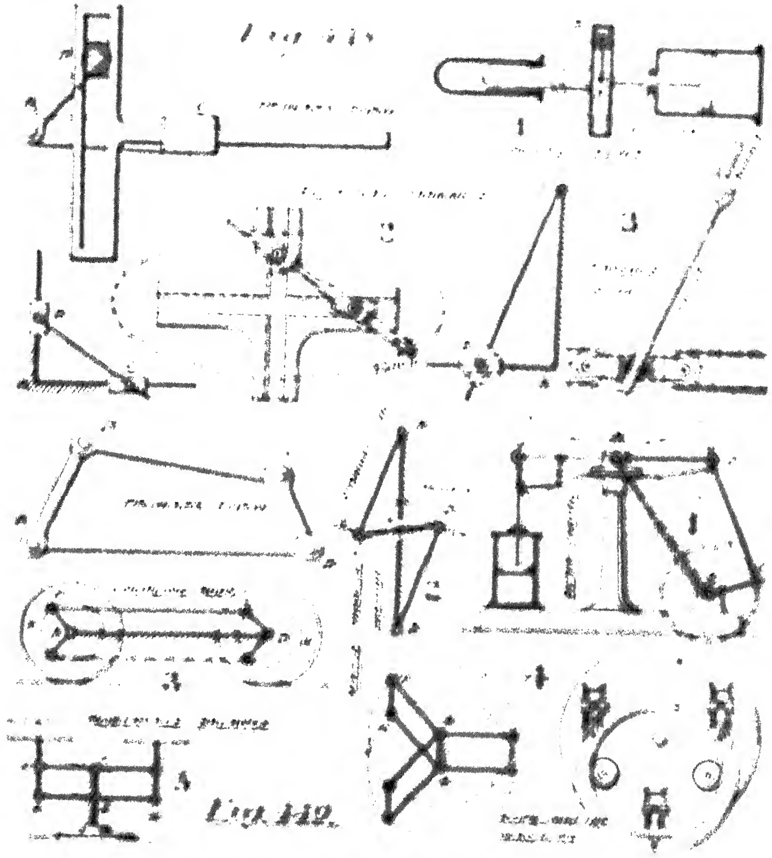


Fig. 119.

Classification of Higher Pairing. — [illegible] of point contact are included as follows:

Rigid Links	Flexible Links
Spur gearing	[illegible]
Bevel gearing [illegible]	
Friction gearing	Pulleys of all kinds with rope, chain, or strap connection.
Cams	
Escapements.	

and these are usually combined with lower pairs.

Flexible links are called *tension elements;* and fluid connections, as between boiler and engine, or accumulator and machine, are termed *pressure elements*, but the latter are always connected to lower pairs. A pump is kinematically the same as a ratchet, the valves being equivalent to pawls (see Fig. 450).

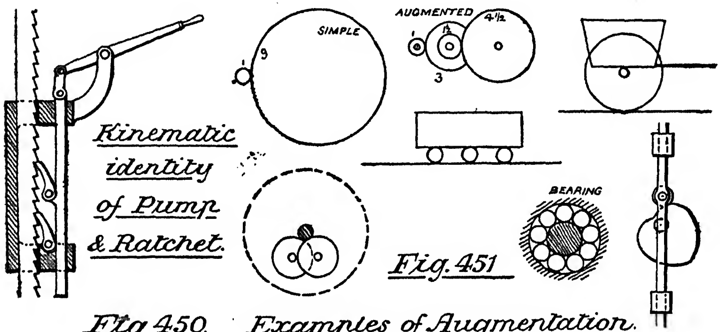


Fig. 451

Fig 450. Examples of Augmentation.

Augmentation of Chains is the multiplication of parts, for convenience or the reduction of friction. Trains of gearing, and anti-friction rollers (Fig. 451), are examples.

Summing up, mechanism may be divided into simple *chains*, formed of rigid or flexible *links*, which are again united by higher or lower *pairs;* and all chains must be *closed*, either by the chain or by external force.*

List of Kinematic Chains.

Lower pairing.	1. Crank chains :	Sliding and turning and screw pairs.
	2. Screw chains :	
High and low pairing.	3. Pulley chains :	Tension and pressure elements.
	4. Wheel chains :	Uniform motion.
	5. Cam chains :	Variable motion.
	6. Ratchet chains :	Intermittent motion.

A *driving* and *working* end are recognised in each of these, corresponding to P and W respectively, and the

Velocity Ratio of P and W in Kinematic Chains will now be investigated graphically. Considering the instan-

* Friction closure is one form of force closure.

taneous motion, in direction only, of the two ends P and W of a link, each point may be supposed, *for the instant*, to be travelling in a separate circle, whose radius will be at right angles to the aforesaid motion, and the two radii will, unless the directions of motion are parallel, meet on one side or other of the line P W. The meeting point is known as the instantaneous or **virtual centre,** and the *ratio of the velocities of P and W will be the same as that of the radii from the virtual centre.* Of course these may change at every instant, and the centre itself will move along a path known as the *centrode.*

Crank and Connecting Rod (Fig. 452).—In the position given, W is travelling tangentially, and W D is its virtual radius, while P is moving towards A, and has a radius P D. D then is the virtual centre, and at the instant considered, the movements being along the dotted arcs, p_1 w_1,

$$\frac{\text{vel. P}}{\text{vel. W}} = \frac{\text{D P}}{\text{D W}}$$

Taking various other positions, we may obtain a series of virtual centres; and through them draw the centrode E D F, where E and M are the positions of P when W crosses the line K E. The curve passes out to infinity at F and O, reappearing at L and Q, the direction being given by the line J P when W is at G and P at H. This means that P and W have then equal velocities. The relative velocities being found for any position, their inversion will give the relation of the forces P and W.

Curve of Velocities.—It is often required to construct a curve of velocities for *one* of the points, *when the other moves uniformly*. Taking the second diagram in Fig. 452, the triangles W C A and W P D are similar, so that

$$\frac{\text{vel. P}}{\text{vel. W}} = \frac{\text{D P}}{\text{D W}} = \frac{\text{A C}}{\text{A W}}$$

Assuming W to move uniformly, being provided with a fly-wheel, A W *will represent crank velocity*, while the projection of P W upon the vertical at C or *c* will give A C or A *c* *the piston velocity*. In Fig. 453 the value A C is found and transferred to the line A W at A E, and this being done for all positions, the ovals or polar curve may be traced, whose *radius vector* always shews P's

velocity for the given position of crank, while the crank arm itself gives W's velocity. Taking various positions of P on H J, and setting up the corresponding polar radii, the curve of P's velocity is obtained as H K J, while the ordinates A W, set up dotted on a base N O of half crank circle circumference, shew crank velocity.

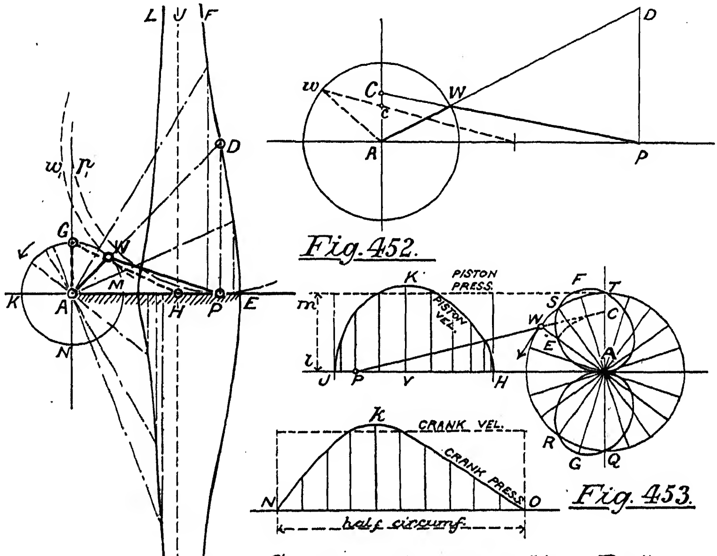


Fig. 452.

Fig. 453.

Crank and Connecting Rod.

Assuming P's pressure as uniform, the ordinates *l m* will give a curve of pressure; and the A E ordinates, being transferred from the polar curve to the base N O,. will give a curve of tangential pressures on crank. Notice points Q R and S T, where P and W have equal velocities, and also points F and W, where P has its highest velocity, and W its greatest pressure.

Time and Distance Bases.—The profile of velocity curve depends on the terms in which we state the base-line divisions. The curves in Fig. 434 are drawn with a *time* base line (equal times), but the oblique lines at C and D would be parabolas if a *distance* base (equal distances) were used. In Fig. 453, H J is a distance base, but supposing N O to represent

piston travel, N k O would be P's velocity on a time base. The ordinates at corresponding times are always the same, but the abscissæ vary, and the two cases must be thoroughly grasped by the student.

Acceleration Curves shew the rate at which the velocity is changing. Let a point move from A to B, Fig. 454, with changing velocity, as shewn by the curve A C B, A B being a *distance base* (here a necessity). Draw any tangent D E F and a normal E G, drop the perpendicular E H, and turn H G round to line H E, giving a point in the acceleration curve. Continuing the construction for various points, K L M is obtained, whose ordinates shew acceleration from A to L, and retardation from L to B.

N.B.—If velocity and distance scales are the same, the acceleration may be measured to the same scale; but, if otherwise, and

v = ft. per sec. of velocity to one inch,
d = ft. distance to one inch,

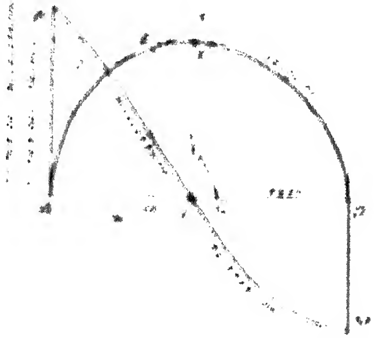
a new acceleration scale must be made, being the velocity scale, stretched or compressed in the ratio $\frac{d}{v}$. (*See Appendix II., p.* 863, *for proof; see also p.* 674.) (*See pp.* 932, 1099, *and* 1106.)

The Oscillating Lever is examined in Fig. 455. The virtual radii are drawn: W B a normal to the circumference, and P B perpendicular to W J. Then:

$$\frac{\text{vel. P}}{\text{vel. W}} = \frac{\text{BP}}{\text{BW}} \quad \text{or as} \quad \frac{\text{AD}}{\text{AW}}$$

For A D being ‖ to J B, the triangles W D A W J B are similar. Turning A D round to A C, we obtain one point in the polar curve, found as at Fig. 456, where A D W is right angle. W's velocity being uniform, the polar radii shew P's velocity. The centrode curve passes to infinity at K, N, G, and P, the direction of the dotted lines being at right angles to W P, when the latter is tangential to the crank circle, namely when P and W have uniform velocities.

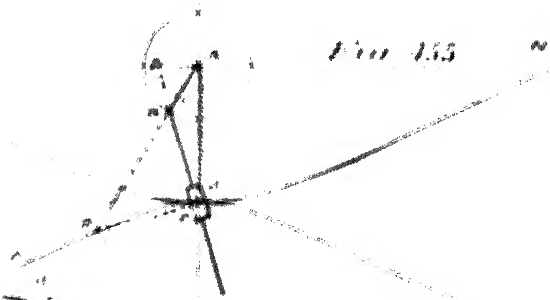
Whitworth's Quick-return Motion (Fig. 457).—B P being the driver, revolving uniformly, the angular and linear velocities of W are to be found. Produce P A to C and P B to D,



Relation of Velocity and Acceleration curves

Fig 154

Fig 155



Oscillating Lever

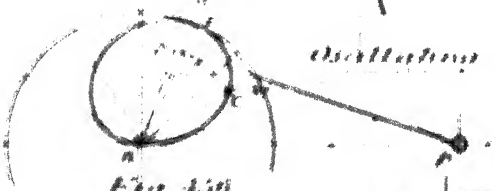


Fig 156

Fig 157

Whitworth's Quick-return

[illegible]

A series of points [illegible] centre [illegible], but the radius [illegible]

[illegible]

Since [illegible] A series of points [illegible] of W_1 measured from centre [illegible] linear velocity. (See Plate XI and fig. [illegible].)

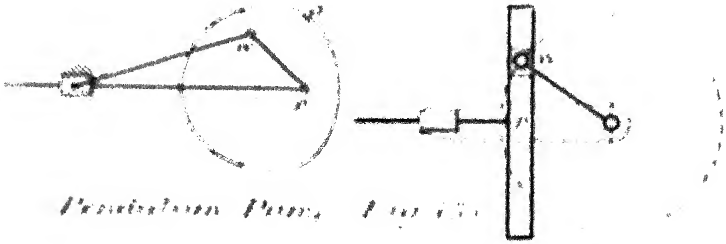
The Pendulum Pump is treated [illegible] at Fig. 458, where the centrode is drawn, and

$$\frac{P\text{'s vel.}}{W\text{'s vel.}} = \frac{[illegible]}{[illegible]} \text{ for any position.}$$

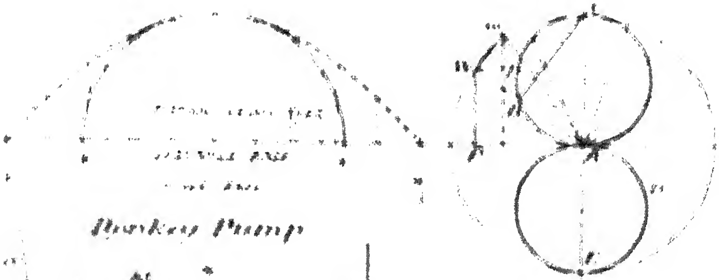
The Donkey Pump (Fig. 459). Taking the [illegible] diagram, we may imagine W moved a small amount tangentially, as W to [illegible], and drop [illegible] perpendicular. Then, velocity being proportional to distance travelled,

$$\frac{\text{vel. } P}{\text{vel. } W} = \frac{p\,w}{W\,w} = \frac{[illegible]}{[illegible]} = \frac{[illegible]}{[illegible]}$$

for the triangles pWw and [illegible] are similar. A series of points, such as [illegible], will define the polar curve, which is a circle, because [illegible] is a right angle (Euclid, III. 31), and while [illegible] shows W's

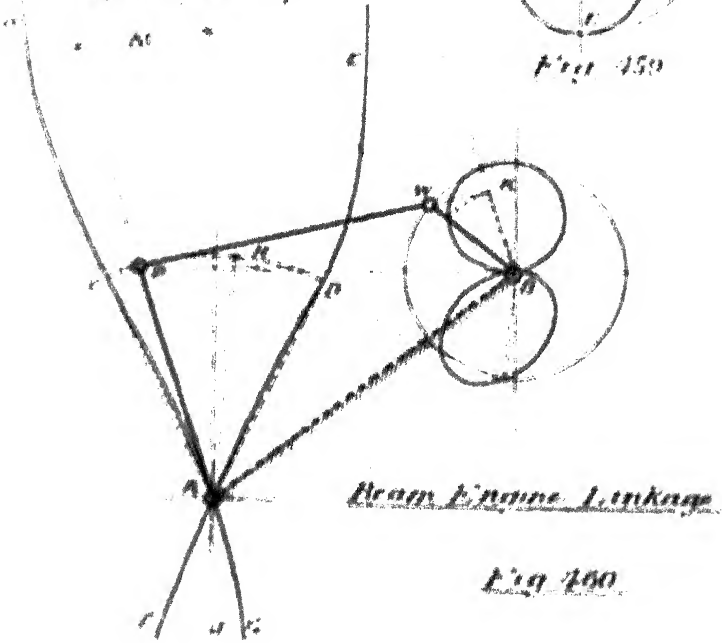


Pendulum Pump *Fig* [illegible]



Donkey Pump

Fig 459



Beam Engine Linkage

Fig 460

velocity, the radius vector shews P's velocity. The motion of P is known as *pure harmonic*, and occurs often in natural science. Transferring P's velocities to a distance base gives a semicircular curve, but on a time base forms the *curve of sines*.

The Beam Engine linkage is shewn in Fig. 460, with centrodes and polar curves. The lines A P, B W, being at right angles to the direction of motion of P and W respectively, will, if produced, give the virtual centre M. Then if B K be || to A P, the triangles M P W and B K W are similar, and

$$\frac{\text{vel. P}}{\text{vel. W}} = \frac{\text{M P}}{\text{M W}} = \frac{\text{B K}}{\text{B W}}$$

the polar curves being completed as before. The centrode curve only reaches infinity on the side J, when A H, B W are parallel; the ends O E meeting at a very great but finite distance. The polar curves are similar to those of the crank and connecting rod, P having greater velocity than W at times. When in the form 3, Fig. 449, the quadric chain has its virtual centres always at infinity, and therefore P and W have like velocities. (*See App. II., p.* 863.)

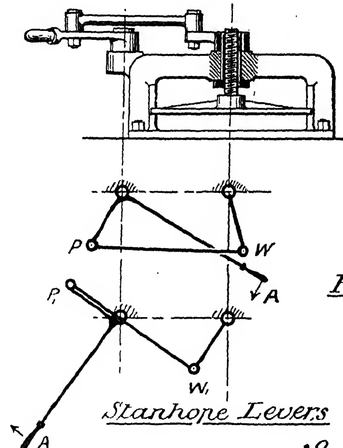
Point paths are often of more importance than forces, but can always be obtained by drawing the links in successive positions; and *the mechanical advantage of a complex system is the product of the advantages of its parts.* Taking now the power transmitters in order,

(1.) **Linkwork** is suitable only for short distances, as in the case of locomotive coupling rods, and is rather a modifier than a transmitter. We shall take a few further examples.

The Stanhope Levers, Fig. 461, were applied by Lord Stanhope to his printing press. Two plan views are given: at first P and W have nearly equal velocities, but when they have moved to the positions P_1 and W_1, the latter has *no velocity*, while the former has yet the original motion.

$$\therefore \frac{\text{P's vel.}}{\text{W's vel.}} = \frac{1}{0} \quad \text{and} \quad \frac{W}{P} = \frac{\text{infinity}}{1}$$

This means that a very great pressure is exerted at W when the paper and type are in contact. A polar curve for W's velocity has been drawn in the right-hand diagram, considering P's velocity



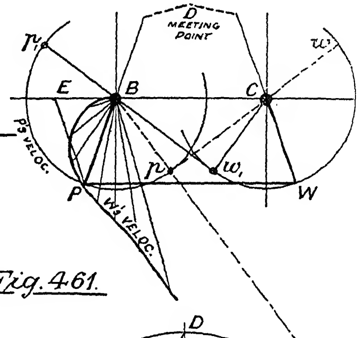
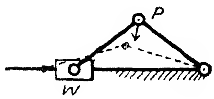


Fig. 461.

Stanhope Levers



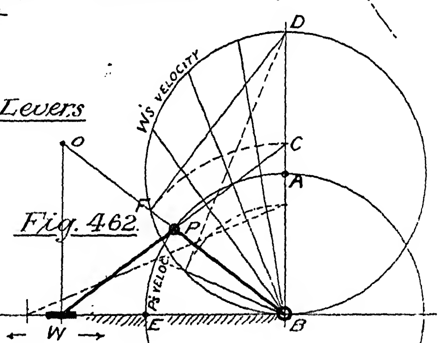
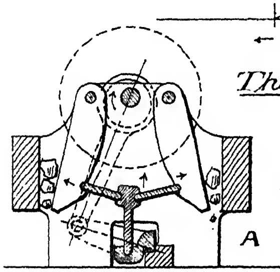


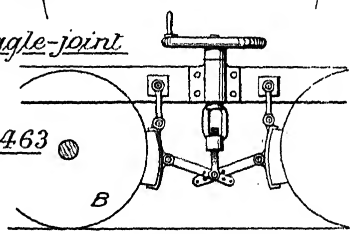
Fig. 462.

The Toggle-joint

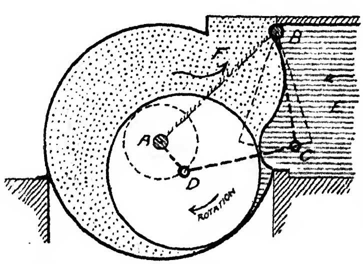


STONE-BREAKER

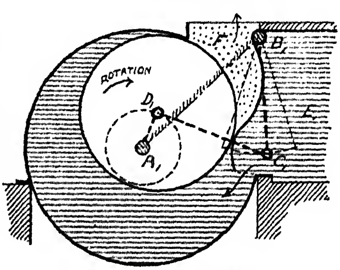
Fig. 463



WAGON BRAKE



1ST POSITION



2ND POSITION

Cooke's Ventilator. Fig. 464.

constant. D is the virtual centre, and D P, D W the radii; and the triangle P E B being similar, P B may represent P's constant velocity, while P E shews that of W. The latter being transferred to B P, gives points in the curve shewn; reaches infinity in the direction B p, and nothing in the direction B p_1. W is then respectively in the positions w and w_1.

The Toggle Joint has many useful applications, the stone-breaker and wagon-brake (Fig. 463) being examples. In Fig. 462 the joint is seen to consist of a simple slider-crank chain. O is the virtual centre, and O P, O W the radii. Producing W P to C,

$$\frac{\text{vel. P}}{\text{vel. W}} = \frac{\text{BP}}{\text{BC}} = \frac{\text{BP}}{\text{BF}}$$

and several points, such as F, will form the polar curve B F D, showing W's velocity, where P's velocity is uniform and represented by B P. The curve is a semicircle, having A as centre.

Cooke's Mine Ventilator in Fig. 464 is a case of the quadric chain. Crank and shutter shafts are connected by link C D, and A B is a fixed though virtual link. Two positions are shown, the shaded air being drawn in, while the dotted air is pushed out by the rotation of the drum.

Quick-Return Motion.—See Fig. 457.

Valve Motion for engines needs examination only for point paths, and will be treated in Chapter X.

Parallel Motions should strictly be termed *straight-line motions*, but are now best known by the first title. Watt's (Fig. 465) is the simplest. A D and B C being equal, the upward movement of P will be vertically straight, because D curves to the left by the same amount as C deviates to the right. This is extremely near the truth when a is below 20°, but not absolutely so. Thus:—

$$\theta^\circ = \frac{2r}{l}(1 - \cos a) \qquad (1)$$

$$\sin \beta = \sin a + \frac{l}{r}(1 - \cos \theta) \qquad (2)$$

$$\left.\begin{array}{c}\text{Deviation of P from}\\ \text{vertical}\end{array}\right\} = \frac{r}{2}(\cos a - \cos \beta) \qquad (3)$$

assuming C D to be vertical at central position.

To use the formula, first find θ, then the angle for sin φ, and [illegible] the deviation, which is really due to a slight inequality between θ and φ. If $l = 12$ and $r = 24$, then when $\theta = 20°$, $\phi = 20°$ [illegible], and the deviation is [illegible], but is practically un-calculable at much below 20°.

Peaucellier's motion, Fig. 466, consists of seven links, and is [illegible]. It may, however, be adapted for [illegible]

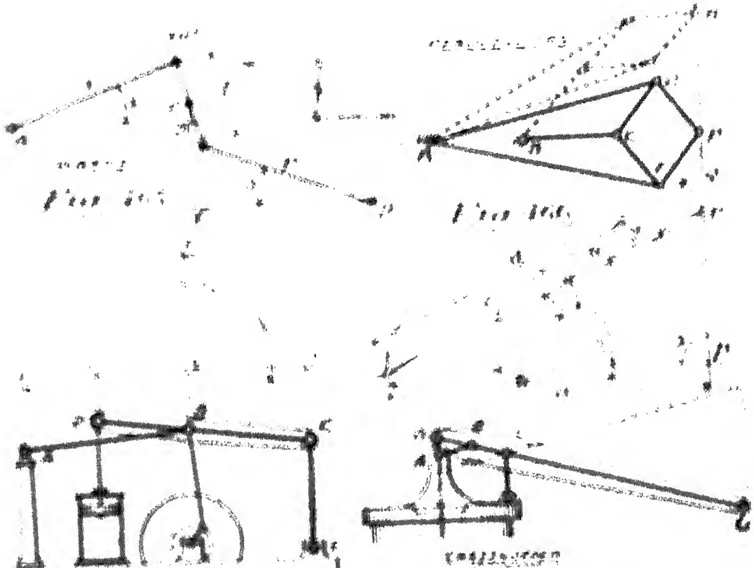


Fig. 465 *Fig. 466* *Fig. 467*

Parallel Motions.

extreme travels, being absolutely correct. P describes the vertical straight line P B, which may be proved geometrically, first proving that A F, F E, E C, and C D are equal, while A B = B C

$$x^2 - y^2 + (z + w)^2 = y^2 + z^2 + 2zw + w^2$$

$$h^2 = y^2 + w^2$$

Subtracting, $x^2 - h^2 = z^2 + 2zw = z(z + 2w) = zd$

This being strictly general, we have, at position p_1,

$$x^2 - h^2 = z_1 d_1 \quad \text{and } z d = z_1 d_1$$

or $z : z_1 :: a_1 : a$, and the triangles are similar, so that angle $\alpha =$ angle β. But α is a right angle, being in a semicircle.

$\therefore$ Angle β is *always* a right angle, and

pr is a straight line.

Scott-Russell's motion 5, Fig. 447, merely copies at A D the truth of the slide C, D A C being always a right angle. A more convenient form is the

Grasshopper motion, Fig. 467, where the slide is replaced by a long link. The gear may be formed (1) with A B = B C = B D as in Fig. 447, or (2) A B : B C : : B C : B D, the second being used in grasshopper engines and the first in a steam crane built by Messrs. R. & W. Hawthorn, where a piston connects directly with D to lift the load. The relation of the links in case (1) may be found graphically: produce points D, B, C, to the respective positions 1, 2, 3, on the base line 1, 3, : with centre 2 strike arcs 1, 4, and 3, 5 : join 4, 5, and draw 5, 6, at right angles to 5, 4. Then 6 produced gives point A, and length of A B; for 5, 2 is a mean proportional between 6, 2 and 2, 4.

The Feathering Paddle-Wheel is shewn in Fig. 468. If the vessel move to the right with a velocity v_s while the wheel rim has a linear velocity of v_f, the floats should enter and leave the water in the directions v_r if they are to meet the water without shock, for v_r is the relative velocity of float to water, found by completing the parallelogram. The controlling mechanism is obtained by quadric chain H G K E where H G is the fixed link.

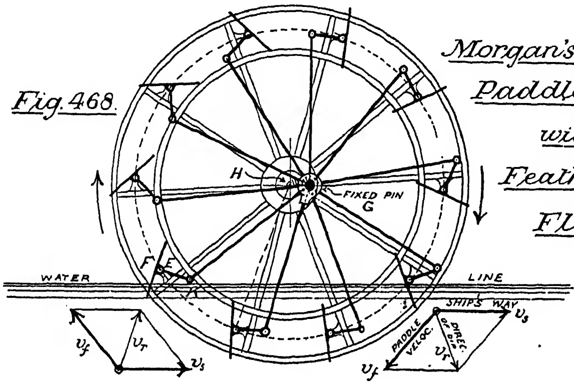
Stresses in Linkwork Members may be ascertained from the principles in Fig. 423 *et seq.*, the structure being balanced by known external forces.

The Work Done at any point of a machine is obtained as at Fig. 325. Taking the case of harmonic motion for donkey pump, let total piston pressure P be uniform during stroke d': then $Pd' =$ work done at P and is shewn by diagram in Fig. 469. Setting out the pressure-curve for W, on a base π R, as explained in Fig. 453, the mean of the ordinates will be found to be ·636 P, and as

$$\text{work put in} = \text{work taken out}$$
$$P \times 2R = \cdot 636\, P \times \pi R$$

which are equal, or no work is either lost or gained in transmission, if friction be neglected.

Fig. 468.



Morgan's Paddle-wheel with Feathering Floats.

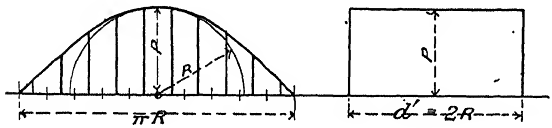


Fig. 469.

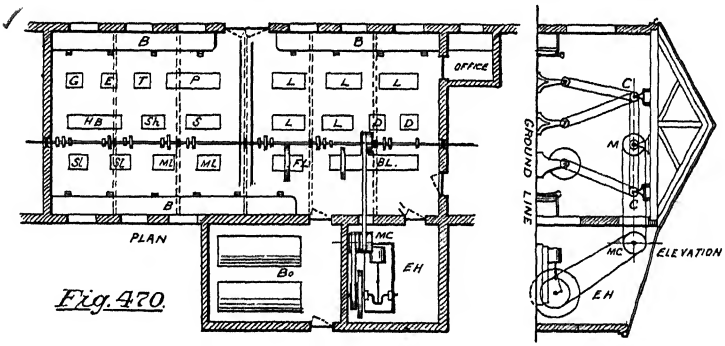


Fig. 470.

Arrangement of Machine Shop.

(2.) **Shafting** is used extensively for power distribution in workshops, being combined with belting and toothed gearing. Fig. 470 is the plan of a small shop as usually arranged. The

[illegible]

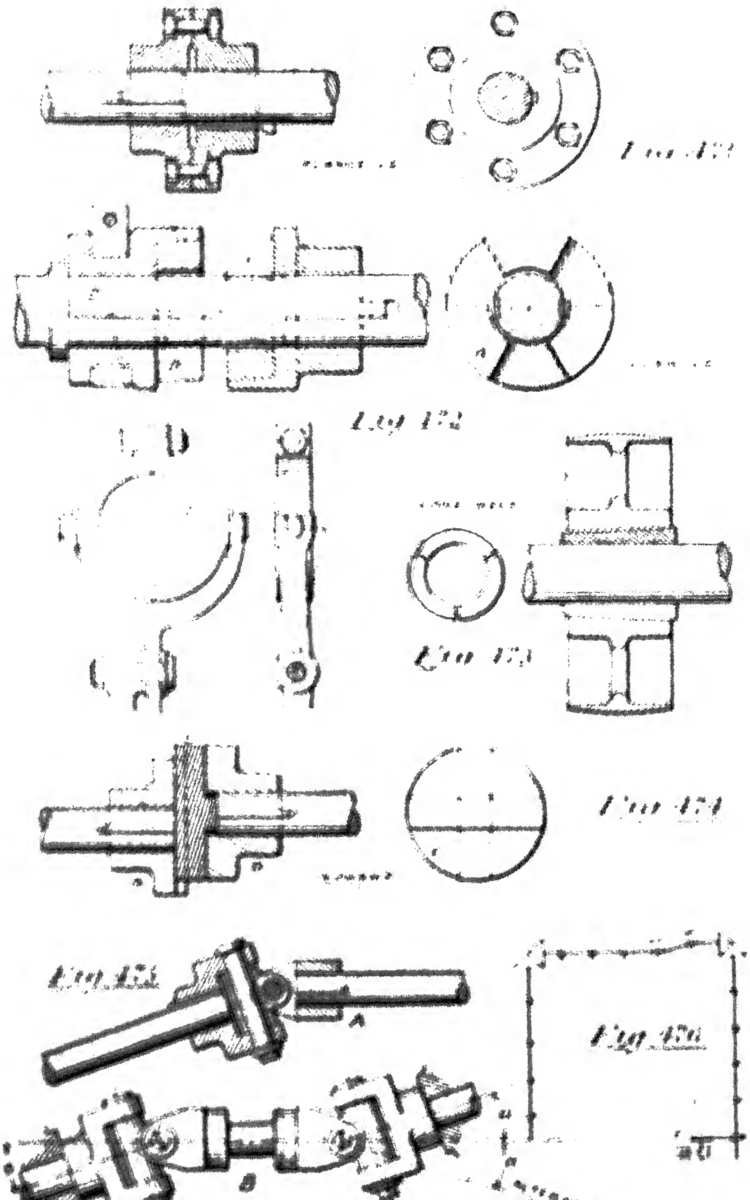
[illegible] a slipping and so shortening [illegible] a grindstone, or an emery wheel, and [illegible] a surface plate. Next, the countershafts are placed [illegible] one for each machine [illegible] power taken first to these and thence to the machine [illegible] the latter to be stopped and started by moving the belt [illegible] without taking the belt [illegible] off the [illegible] pulleys should be placed as near bearings as possible [illegible] balanced to avoid vibration. Where [illegible] the countershaft [illegible] on the [illegible] decrease in diameter, when further [illegible] arrange [illegible] changing pulley positions [illegible] of average small machines (about ½ H.P. [illegible]) the pulley is simply gripped tight on the shaft [illegible] to Whitworth gauge; the main pulley [illegible].

Naturally, considerable power is required [illegible] the shaft without further transmission [illegible] per cent. of the total power required [illegible] *Mag.* and *App.* [illegible]

[illegible] are fitted [illegible] bearings, and [illegible]

Couplings connect the separate lengths of shafting, the most usual form being the flange coupling, Fig. [illegible], consisting of two discs, one keyed to each shaft, and both bolted together, the bolt heads being sunk for safety. For strength, see Fig. [illegible]. Couplings should be placed near bearings, and sometimes serve as pulleys. Marine shafts have the discs forged on to reduce weight, and thus form *solid* couplings (see Plate XVII., facing p. [illegible]).

Clutches unite shafts, or pulleys and shafts, so as to admit of disengagement when required. Fig. [illegible] shows the common



Couplings

'claw' clutch. C is fixed by key to the right-hand shaft, and B slides on a pair of feather keys D in the left-hand shaft, so that the claws at A may be locked or unlocked. The clutch strap E encircles the clutch B, and is further grasped by the fork lever: this gives a sufficiency of wearing surface between the rotating clutch and stationary lever. The difficulty of entering the jaws is met by the adoption of friction clutches. (*See pp.* 569–70.)

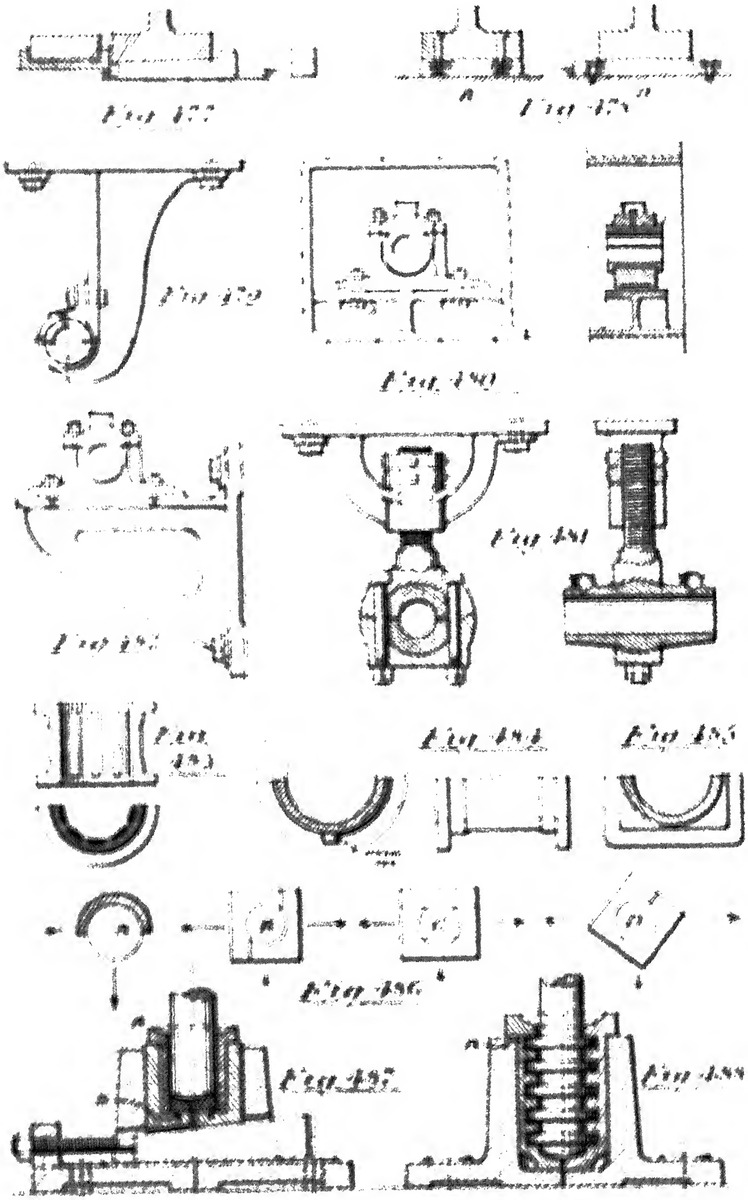
Two shafts slightly out of line but mutually parallel may be united by the *Oldham coupling*, Fig. 474. A middle plate C, having cross strips, unites with grooves in the flanges A and B, and the velocity is transmitted unimpaired. If the shafts are mutually inclined, the *Hooke's or Universal Joint*, A, Fig. 475, must be employed, and if considerably out of line though parallel, B must be used. A transmits the velocity unevenly, but the double arrangement B rights this difficulty. Fig. 476 was adopted for many years at a northern establishment: E is the engine, and U J are universal joints, while the three shafts represent three separate shops. (*See Appendices I. and II., pp.* 763 *and* 866.)

Keys were examined in Figs. 374–5. The sunk key is best, but the flat key is more often used in shop shafting. *Cone Keys* (Fig. 473) are made from a hollow cone, turned and afterwards divided: they give a very perfect grip.

Keys should have a taper in depth from front to rear, and a gib-head adopted as in Fig. 477, if there are no means of otherwise releasing the key. Although some workmen fit keys at top and bottom only, they should no doubt fit accurately both at top and sides. Shrinking boss on shaft gives very great security. Keys are sometimes forged on the shaft. (*See p.* 423.)

Feather or sliding keys can be fastened either to boss or shaft as most convenient. See A and B, Fig. 478.

Bearings are strictly gun-metal supports termed bushes, but the supporting brackets take various forms. Fig. 479 is a common hanger, Fig. 480 a wall box, and Fig. 482 a wall bracket. The last two have bearing and bracket separate to allow of adjustment. Fig. 481 shews a special hanger, having a long cast-iron bearing lying in a spherical seat which adjusts itself automatically to the shaft deviation. Permanent vertical adjustment is obtained by screw and nut.



Bearings

[illegible]

[illegible] *Thrust Bearings* [illegible] *Figs. 177, 178* [illegible] In the example there are [illegible] rings [illegible] wear [illegible] as much as [illegible] *Kent and Hays*.

Bushes are [illegible] brass, and in halves as at Fig. 178 [illegible] being planed on the meeting edges, they may then be [illegible] together, turned, and afterwards split. If made separate [illegible] 178, they are planed throughout. [illegible] depends on direction of pull. Fig. 178 [illegible] as at *a* [illegible] and *c*, and a horizontal engine [illegible]. If one [illegible] can be used, the oiling [illegible] Fig. 178 shows grooves filled with white metal [illegible] with large shafts having variable [illegible] difficult, the particles of soft metal [illegible] the surface [illegible] a [illegible] and to prevent seizing.

Journals on shafting are often [illegible] they are formed either by turning down [illegible] collars as at *b* (Fig. 179). The allowable [illegible] pressure

Journals Fig. 179.

is reckoned on the projected area $l \times d$, and varies very much with the speed of the journal surface, being low enough to avoid squeezing out the oil. High-speed shafts have their journals made as small as strength will permit, while the surface is obtained by increased length, and the work lost in friction thereby reduced.

but in slow speed shafts the frictional loss depends very little on the speed, and the journal diameters are therefore large. The following very useful table is taken from 'Unwin's Machine Design.'

ALLOWABLE PRESSURE ON PROJECTED AREA OF JOURNALS.

Purpose.	Pressure in lbs. per sq. in.
Very slow speed journals	3000
Cross-head journals	1200
Crank pins for slow engines	800 to 900
Marine crank pins	400 to 500
Marine crank bearings	400 to 600
Railway journals	300
Crank pins for small engines	150 to 200
Marine slide blocks	100
Stationary engine slide block	30 to 60
Propeller thrust bearings	50 to 70
Main shafting in cast iron bushes (Sellers)	15

The ratio of l to d must next be decided by the following empirical formula

$$\frac{l}{d} = \cdot 003N + 1$$

which agrees well with practice. For the journal in Fig. 481, at 1000 revs. per m., the ratio is 4 : 1.

Pressure on Pivots or footsteps should not exceed 250 lbs. per sq. in., while that on *Collar Bearings* may be between 15 and 90 lbs., being at 50 lbs. for thrust bearings of steamships.

(*See p.* 871.)

Horse-power Transmitted by Shafting.—Taking a round shaft, let w be applied to the end of a 12" arm.

$$w \times 12 = \frac{f^{\text{lbs}}\pi d^3}{16} \quad \text{and } w = \frac{f^{\text{lbs}}\pi d^3}{12 \times 16}$$

w being exerted through 2π feet at every revolution:

$$\text{H.P.} = \frac{w \times 2\pi \text{N}}{33000} = \frac{f^{\text{lbs}} \pi d^3}{12 \times 16} \times \frac{2\pi \text{N}}{33000} = \frac{f^{\text{lbs}} d^3 \text{N}}{320810}$$

$$\text{and } d = \sqrt[3]{320810} \sqrt[3]{\frac{\text{H.P.}}{f^{\text{lbs}} \text{N}}} = 68{\cdot}44 \sqrt[3]{\frac{\text{H.P.}}{f^{\text{lbs}} \text{N}}}$$

$$\therefore d \propto \frac{\sqrt[3]{\text{H. P.}}}{\sqrt[3]{f} \sqrt[3]{\text{N}}}$$

Example 47.—A shaft transmits 20 H.P. at 100 revs. Find (1) how many H.P. it will transmit at 250 revs., and (2) dia. to transmit 40 H. P. at 250 revs. with f at 2 tons per sq. in. for stiffness.

(1) $$\text{H. P.} \propto d^3 \text{N}$$

$$\therefore \quad 20 \propto 100 \quad \text{and} \quad \text{H. P. req.} \propto 250$$

$$100 : 20 :: 250 : \text{H. P.} \quad \underline{\text{and H. P.} = 50}$$

(2), $$\underline{d} = 68{\cdot}44 \sqrt[3]{\frac{40}{2 \times 2240 \times 250}} = \underline{2{\cdot}25''}$$

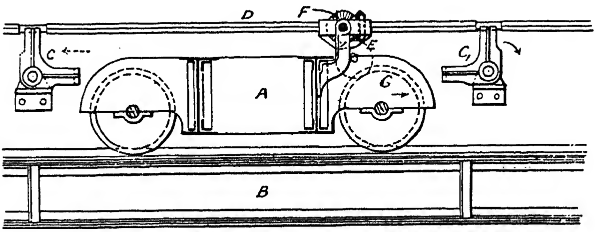
Example 48.—Compare the weight of shafting in a twin with that in a single screw ship, neglecting couplings: the H.P. in each being the same and the speed of each twin being 25 °/ₒ above that of the single screw. (Hons. Mach. Constr. Ex., 1886.)

$$d \propto \sqrt[3]{\frac{\text{H. P.}}{\text{N}}} \ldots \propto 1 \text{ for single shaft} \ldots \propto \quad {\cdot}73 \text{ for each twin shaft.}$$

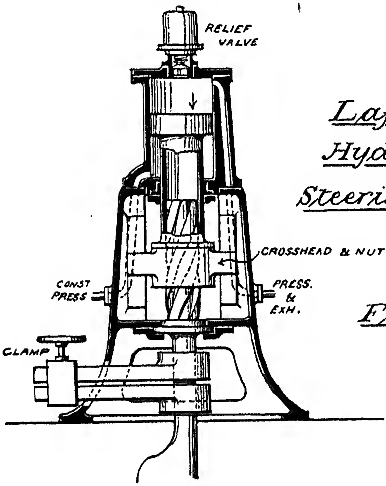
$$\underline{\text{Weights} \propto d^2} \ldots \propto \underline{1 \text{ for single shaft}} \ldots \propto \begin{cases} {\cdot}73^2 \times 2 \quad \text{or as} \\ \underline{1{\cdot}066 \text{ for the two screws.}} \end{cases}$$

Square Shafts are often adopted in travelling cranes. In Fig. 490, B is the longitudinal and A the cross girder of a crane, the power being given from shaft D through mitre gear to F, and by spur gear to G. As the carriage moves along B, the tumbler bearings are turned through a right angle, and are only off the shaft during the passage of the mitre wheels, the bracket at E being shaped to serve as a tappet.

Long screws sometimes serve as shafts, as in large planing machines with travelling tool, and a linear advance of the screw may produce rotation if sufficiently large in pitch, as in Fig. 491.



Tumbler Bearings. Fig. 490.



Lafargue's Hydraulic Steering Gear.

Fig. 491.

(3.) **Spur Gearing** transmits power between parallel shafts only. Spur wheels are the equivalent of friction discs, having teeth provided to avoid slipping with heavy loads. The teeth are formed partly above and partly below the disc outline, the latter becoming virtual only, and being then termed the pitch line. Thus,

Pitch Circle, Line, or Surface *of a spur wheel or* ... *represents the contour of the ideal disc or straight-edge which* ... *transmit the same motion.*

To transmit perfectly uniform motion the teeth must b specially formed, and all teeth in gear at once must contribu to the perfection of the motion. To fulfil these conditions th normals to all surfaces of contact must pass through the meetir point of the pitch lines (Fig. 492), and this is obtained when or tooth *b c*, on A, is the *envelope* of the relative positions of the oth tooth on B (Fig. 493) when the discs are rolled together. Th teeth are actually drawn, however, in a somewhat differe manner. (*See Appendix III., p.* 926.)

Cycloidal Curves.—A *cycloid* may be traced by a poi on the rim of a disc which rolls along a straight edge, and *epi-cycloid* when the disc rolls upon a circular arc (Fig. 49 A *hypo-cycloid* is similarly traced *within* an annular disc at Fig. 495, noting that when the rolling disc is half th diameter of the annulus a straight line is obtained, as shev dotted; a fact which has produced White's parallel moti (Fig. 496).

Rolling Circle.—The above curves will serve for whe teeth, if the same rolling circle be adopted for parts that cor in contact, the tooth point being formed by epi-cycloids a the root by hypo-cycloids. Taking the wheels A and B, Fig. 49 a rolling circle is first to be chosen as governed by the ro curves: thus, if the circle be half the pitch diameter, radial tee are formed, as at C; if larger, the root will be undercut as at and E is drawn with a circle of $\frac{1}{4}$-pitch diameter. The latt is reasonable, as giving strength, while yet avoiding obliq pressure on bearings. Adopting then the rolling circles shew F may roll the root of B and the point of A, because these are engage, but G will serve for root of A and point of B. When the wheels of a train are to work together interchangeably, t same rolling circle must be used throughout. If the too pressure is always in one direction, as in Fig. 499, a large rolli circle may be adopted for the acting surfaces and a small one the back surfaces, thus giving great root strength without obliq action.

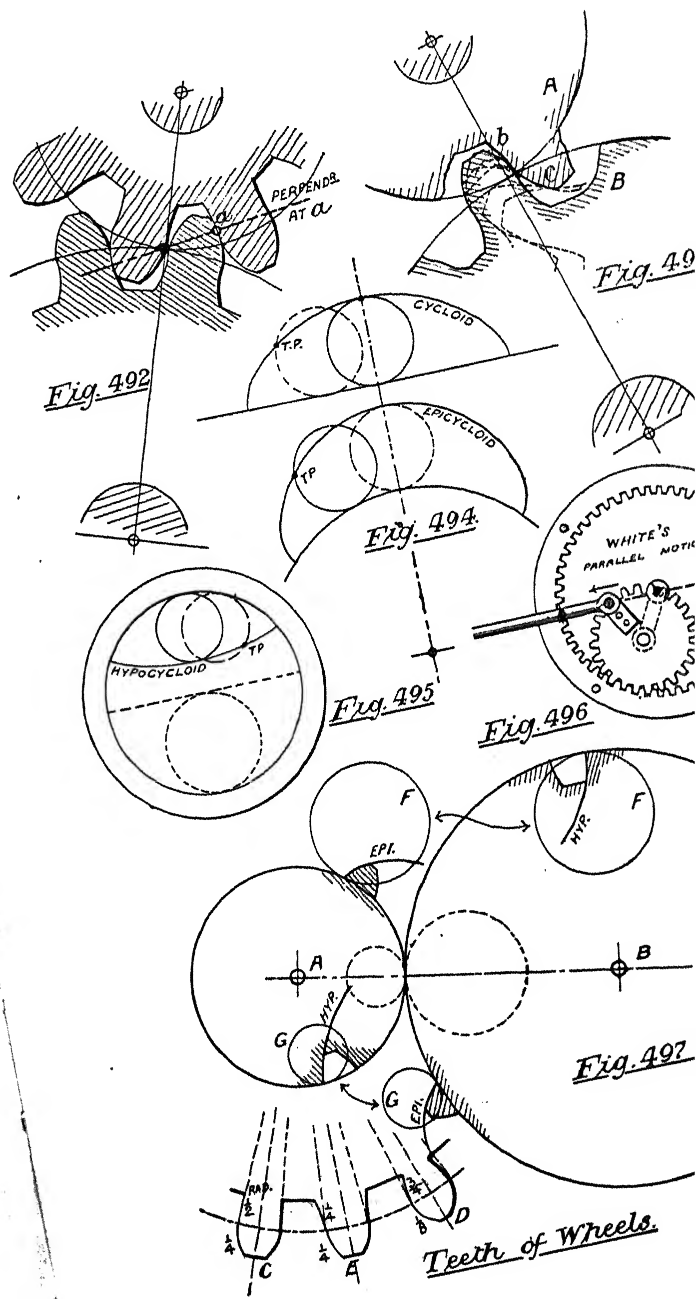


Fig. 492

Fig. 49

Fig. 494.

Fig. 495

Fig. 496

Fig. 497

Teeth of Wheels.

Rolling a Tooth. Referring to Fig. [illegible], [illegible] the pitch line, and [illegible] the rolling circle. As the latter rolls from [illegible] to [illegible] it takes up the various dotted positions, and the tracing point passes to I, II, III, and *b* successively, the positions being found by making *a* I = [illegible], *a* II = [illegible], *a* III = [illegible], and *a* *b* = [illegible], in each case *measured round the curves* by stepping off with dividers. The tooth point being then sketched through, the root curve may be treated in like manner, and the dotted tooth formed by proportions found at a later page. (*See also p.* [illegible].)

For a rack the same rolling circle is used for points and roots, the curves being, of course, cycloids.

Rules for Small Pinions. The ratio of wheel to pinion diameter should not exceed about 8 to 1, or the obliquity of action is great; and the number of the teeth in the pinion should not, if possible, be less than 10, though [illegible] and even [illegible] have been used in extreme cases. If the pinion be double shrouded as at A, Fig. [illegible], the strength is doubled, and wear, which is very great on the pinion teeth, well provided against. Single shrouding as at B is of little advantage.

Arc of Contact. In Fig. [illegible], *a* is the driver, *b* the follower, and C the rolling circle, having tracing points [illegible] upon its circumference. Rolling C within *a*, the hypocycloids [illegible] are described, and the epicycloids [illegible] formed round *b*. But while C touches P, [illegible] are equally ready to describe the one or other set of curves, which means that [illegible] and [illegible] are the only points of contact of curves [illegible] and [illegible] respectively; and all cycloidal curves drawn by C must have their contact points along the arc [illegible]. Supposing *a* to be moved round in the direction of the arrow, the teeth will first touch at [illegible] where *b*'s point crosses circle [illegible]; before this there would be backlash. If [illegible] be struck below P, M shows the last touching point where *a*'s point crosses circle C_3. The path or *arc of contact* will be [illegible], [illegible] being termed the *arc of approach* and [illegible] the *arc of recess*. [illegible] is rather shorter than [illegible], so [illegible] is slightly greater than [illegible], representing the greatest *angles of obliquity* at recess and approach respectively. If these be less than the friction angle,* there will

* The angle whose tan is the coefficient of friction μ. For rough cast iron μ = [illegible], and friction angle = [illegible]°.

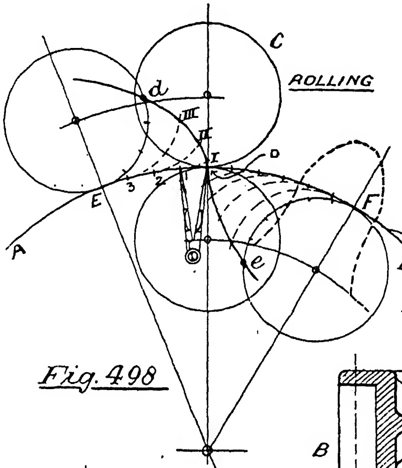


Fig. 498

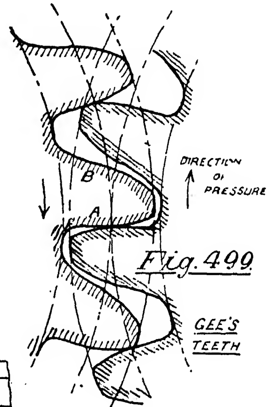


Fig. 499

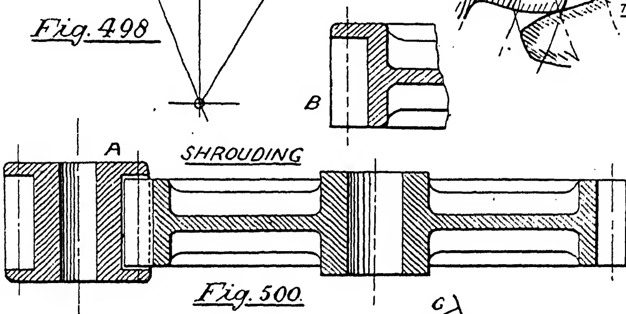


Fig. 500

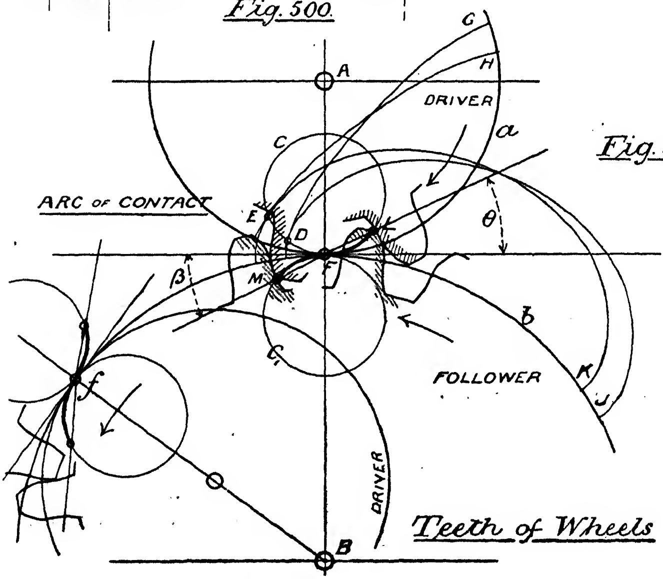


Fig. 501

Teeth of Wheels

to the pressure in direction a c, the latter depending on the difference of β and the friction angle. Drawing the teeth in position at first and last contact, their paths on their respective pitch lines define the *arc of action*, which should be long enough to engage two pairs of teeth at once, and avoid jerks. (*See App. I.*, *p.* [illegible])

Internal or annular wheels are examined in the same manner. The obliquity is somewhat greater on the inside, as at *f*, Fig. 501, and the curves are reversed for the wheel, an epicycloid forming the root and a hypocycloid the point. Tooth point is sometimes called ***addendum***, and ***flank*** used instead of 'root.'

Proportions of Wheel-teeth, as at present adopted, are given in Fig. 502. It is now proposed that they should be somewhat decreased in height, but the objection then is that fewer than two pairs of teeth *may* only be in contact. The pitch *p* should always be measured along the curve of the pitch line. The difference (.52 – .48)*p* is termed *backlash*, and (.4 – .3)*p* is called *clearance*. The former is sometimes eliminated entirely, as in sighting gear for turret guns.

Example 49. Determine the arc of action, and the greatest obliquity of the line of action, in a pair of cycloidal teeth. State also how many teeth are in gear at once when *p* = 2", T = [illegible] and [illegible], dia. of rolling circle = 8⅝", height of points or addenda = [illegible]. (Hons. Mach. Constr. Ex., 1893.)

Fig. 503 is drawn to scale. The arc of contact is from *a* to *b*, and the

arcs of action are shown by radial bounding lines.

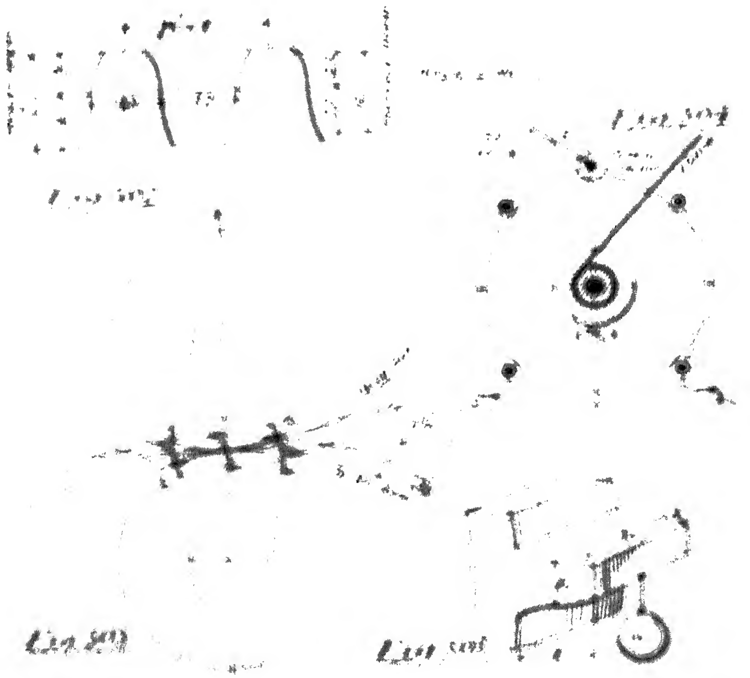
Greatest obliquity = [illegible]°.

There are three pairs of teeth in gear at once.

The latter is found by stepping the pitch into the arc of action. Then number of teeth in gear = no. of *integral* pitches + 1.

Strength of Teeth. The first datum required is the pressure on the teeth.

Example 50. A crab is required to raise ½ a ton by the strength of one man, 30 lbs. on a 15" handle. Sketch the gearing and chain barrel, pinion having 12 teeth of 1¼ ins. pitch, and chain barrel being 7½" dia. Find also pressure on wheel teeth. (Hons. Mach. Constr. Ex., 1887.)



Assuming a possible pitch, the tooth is considered as a cantilever with concentrated load, as at Fig. 505. Breadth b varies some

what in different cases, but $2\frac{1}{2}p$ is a good working value [illegible] is measured at the pitch line. If one tooth bears the whole pressure and [illegible] $= 4 p$ [illegible] $= 2\frac{1}{2}$ tons for cast iron ($f_t = 1$ ton or less where much shock). Then

Safe load on cast iron teeth $= \frac{f_t\, b h^2}{6 l} = \frac{[illegible] \times 4^2 p^2 \times 4^2 p}{6 \times [illegible] p} = \underline{1\frac{1}{3}p^2 \text{ tons}}$

Load may also be estimated in terms of the H.P. transmitted. Thus

$$\frac{w \times 2\pi\, \mathrm{R}\, \mathrm{N}}{33000} = \mathrm{H.P.} \quad \text{and } \mathrm{W} = \frac{[illegible]\ \mathrm{H.P.}}{\mathrm{R}\, \mathrm{N}} \text{ tons}$$

$$\text{But R} = \frac{p\, \mathrm{T}}{2\pi} \qquad \text{Load on teeth} = [illegible]\, \frac{\mathrm{H.P.}}{p\, \mathrm{T}\, \mathrm{N}} \text{ tons}$$

Example [illegible].—A [illegible] toothed wheel [illegible] makes [illegible] per min., transmitting [illegible] H.P. Find pressure on teeth, and pitch when width is [illegible]. (Eng. [illegible], 1892.)

$$\underline{\text{Pressure on teeth}} = \frac{[illegible]}{[illegible]} = [illegible] \text{ tons}$$

$$\text{Safe load} = \frac{[illegible] \times 4^2 p^2}{6 \times [illegible] p} = [illegible]\, p \text{ tons}$$

$$[illegible]\, p = [illegible] \quad \text{and pitch} = [illegible]$$

Example [illegible].—A spur wheel [illegible] pitch and [illegible] face transmits [illegible] H.P. with pitch line velocity of [illegible] ft. per sec. Find H.P. transmitted by a wheel of [illegible] pitch and [illegible] face, the velocity being [illegible] ft. per sec. (Hons. Mach. Constr. Ex., 1881.)

$$w = \frac{f_t\, b h^2}{6 l} = \frac{[illegible] \times 4^2 p^2}{6 \times [illegible] p} = [illegible]\, f_t\, bp$$

$$\mathrm{H.P.} = \frac{w \times \text{face}}{[illegible]} = \frac{[illegible]\, f_t\, bp \times \text{face}}{[illegible]} = [illegible]\, f_t\, bp\, v$$

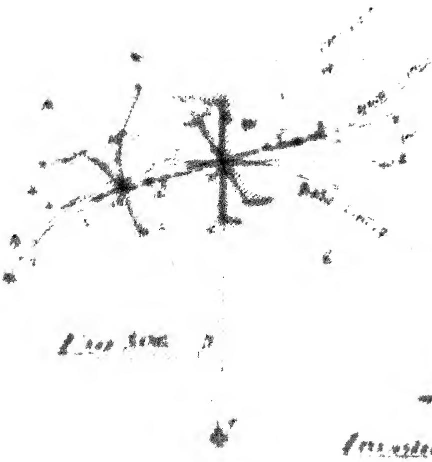
$$\mathrm{H.P.} \propto bp\, v$$

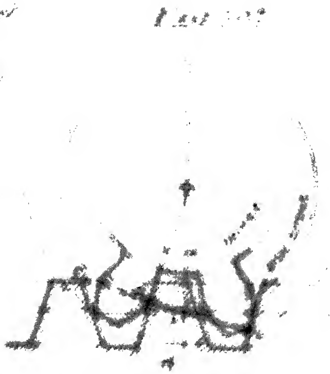
$$(1) \quad [illegible] \propto [illegible] = [illegible]$$

$$(2) \quad \mathrm{H.P.} \propto [illegible] = [illegible]$$

$$[illegible] = [illegible] \quad \mathrm{H.P.} \quad \text{and } \mathrm{H.P.} = [illegible]$$

[illegible]





Involute Teeth

Involute Teeth possess the advantage that their wheel centres may be placed slightly nearer or further apart without disturbing the accuracy of contact. The obliquity is, however, greater than for cycloidal teeth. Fig. [illegible] shows the method of drawing the curves. Draw two circles x and y whose radii are

rack 968 of the [illegible] the path of [illegible] and $a-e$ [illegible] commencement, and at [illegible] the end of contact. [illegible] be fastened at [illegible] and a pencil attached to [illegible] the [illegible] cause the pencil to describe the [illegible] and [illegible] The curve is best found by drawing a [illegible] on tracing paper and [illegible] it round [illegible] without slipping.

Internal teeth are similarly drawn, but the rack, Fig. [illegible], has a base circle of infinite radius ∞, so the teeth curves are straight lines whose angle to line of centres $= a - e - b$.

Safe Velocity of Toothed Gearing, at pitch line, varies from 1800 to 3000 or 4000 ft. per m., the former for rough cast iron, and the latter for machine cut wheels.

Mortice Teeth, Fig. 508, are now little employed. They were introduced to decrease noise and jar, the teeth being wood in one wheel, while the fellow wheel has iron teeth [illegible] filed up.

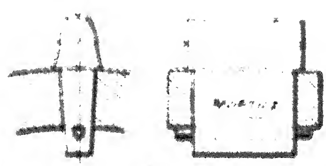
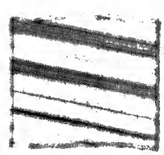


Fig. 508

Fig. 509



SINGLE HELICAL

Fig. 510

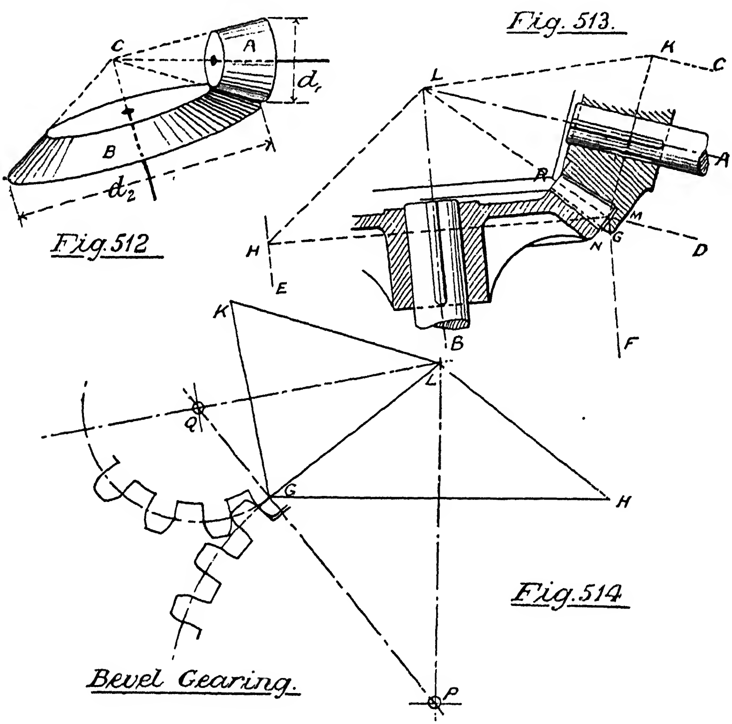
DOUBLE HELICAL

Fig. 511

Helical Teeth, &c. The smoothest action being observed to occur when a very small pitch was used, Dr. Hooke invented his *stepped gearing* as in Fig. 509, to obtain strength and smooth action at once. These were changed later to the form at Fig. 510, for facility in casting and cutting, and recently the *double helical* teeth in Fig. 511 have been adopted to avoid endlong pressure on the bearings caused by single helical teeth. They

are machine-moulded by two half-patterns, and work smoothly if well formed; being said to be stronger than ordinary teeth, which is doubtful.

(4.) **Bevel Gearing** connects shafts whose directions meet at any angle. Their ideal form is that of the frustra of cones, as A and B, Fig. 512, having a common vertex, as C. The pitch diameters are measured at d_1, d_2.



Two shafts A and B, Fig. 513, are to be connected so that their revolutions shall be as 2 : 1. Assume any convenient diameter C D and draw C K and D M ∥ to A L. Taking E F = 2 C D, draw E H and F G ∥ to B L. Through G draw G H at right angles to B L, and G K at right angles to A L: then join H, G, and K to L.

Upon these cones the teeth are [illegible] radiating from [illegible]

Equal bevel wheels with shafts at right angles are termed *mitre wheels*.

Bevel wheel Teeth are set out as in Fig. [illegible], [illegible] being the cones, drawn [illegible] at right angles to [illegible], and with centre [illegible] and [illegible] strike arcs, upon which the teeth are to be designed as though they were spur wheels. But although the teeth are set out at a their strength must be reckoned at b, for there the teeth are weaker in proportion to load than at a. Refer also to pp. 62 and 25[illegible].

(5.) **Worm Gearing** gives large mechanical advantage with few parts. Friction, however, causes considerable loss unless the gear be exceedingly well made. The methods of practical construction are given at pp. [illegible] and 274, the latter being of course preferable. In common with other gear giving high ratio [illegible] with few parts, *e.g.*, Weston block, &c., worm gear possesses the property of **non-reversibility**; **the wheel will not drive the worm unless the pitch be excessive**. The reason is that the direction of pressure is within the friction angle and W is placed at a disadvantage.

$$\text{Mech. Adv.}\ \frac{W}{P} = \frac{\text{No. of threads in worm wheel}}{\text{No. of threads on worm}}$$

Usually the denominator is unity. Plate VII. and Fig. [illegible] give good examples. For the latter

$$\left.\begin{array}{r}\text{Total Mech. Adv.}\\ \text{neglecting friction}\end{array}\right\} = \text{Adv. of worm} \times \text{Adv. of screw}$$

$$= \frac{46}{1} \times \frac{[illegible]}{[illegible]} = \frac{[illegible]}{1}$$

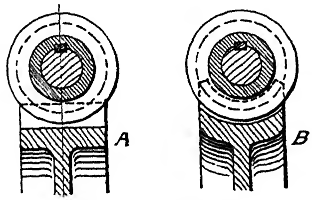
$$\left.\begin{array}{r}\text{Man's pull on handle}\\ \text{when 30 tons are}\\ \text{on jack (neglecting}\\ \text{friction)}\end{array}\right\} = P = \frac{W}{[illegible]} = \frac{[illegible]}{[illegible]} = \underline{20\ \text{lbs. nearly.}}$$

(*See Appendix IV.*, § [illegible])

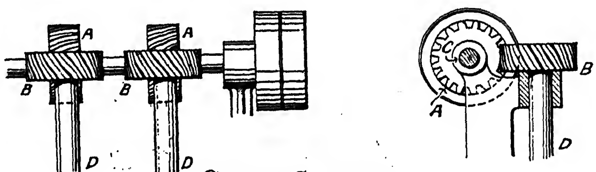
Fig. 515 shews the forms of teeth, a being the best, though b serves well enough for light pressures.

Screw Gear is used to connect shafts that do not intersect, when moderate ratios are required. It is really exaggerated worm

gear, with so many threads to the worm that it becomes a wheel. Fig. 516 shews its application in a Multiple Drill where A A are the drivers, and B B follow on the drill spindles D D. The wheels are here equal, and the teeth are inclined at 45° to the axis.



Worm Gearing.
Fig. 515



Screw Gearing
Fig. 516.

Epicyclic Wheel Trains, like worm gear, produce a high ratio with few parts. Kinematically they are ordinary trains where one wheel is the fixed link.

Case I.—Fig. 517. Let A and L be in gear, with A L fixed. If A make a minus rev. with relation to A L, L will have made $\frac{A}{L}$ plus revs., because $\frac{A}{L}$ is the ratio of the train. Next fix A and put L out of gear. If now arm A L make one plus rotation two things have happened: A has made one minus rev. relatively to A L, and L has made one plus rev. relatively to A. Finally, put A and L in gear, and give A L one plus rotation. L receives two motions: *one* plus rev. due to its connection with A L, and $\frac{A}{L}$ plus revs. due to the relative minus turn of A —— both relatively to A, and

$$\text{L's revs.} = 1 + \frac{A}{L}$$

Case II. Fig. [illegible]. The [illegible] motion due to A [illegible] reversed, so that

$$I\text{'s revs} = 1 - \frac{A}{I}.$$

A special case is when A = I, and I's revs = 0, the upright arm always preserving its vertical position.

Fig. 317.

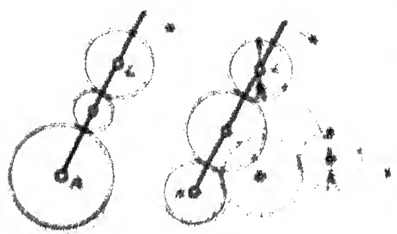


Fig. 318.

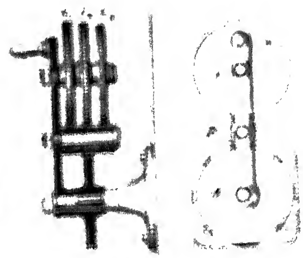


Fig. 319.

In Fig. 319 Ferguson's paradox illustrates Case II., giving three different motions on one axis. Here I_2 has equal teeth with A, I_1 has one tooth more, and I_3 one less. Therefore

$$I_1\text{'s revs} = 1 - \frac{A}{A+1} \text{ and are plus}$$

$$I_2\text{'s revs} = 1 - \frac{A}{A} \text{ and are nothing}$$

$$I_3\text{'s revs} = 1 - \frac{A}{A-1} \text{ and are minus}$$

Case III. Fig. [illegible]. Let A and I be equal. Then, by formula

$$I\text{'s revs} = 1 + \frac{1}{1} = 2$$

relatively to A. We may vary the experiment by carrying A round L, but so that A *does not revolve;* then the relative positions will still be the same, as shewn by a comparison of the figures, and L will again make two revolutions while A is carried once round it.

Watt's sun and planet gear, Fig. 521, is a practical example A slight deviation from the rigid vertical occurs at c and d, but the total result remains; S makes two revs. for one rev. of the crank.

Case IV.—A *Reverted Train* is where A and L turn on the same axis. In Fig. 522, A is fixed and L reverted, while $a\ b\ c$ shews the train in direct order.

$$\text{The train ratio is } \frac{a \times c}{b \times l} \text{ and}$$

$$l\text{'s revs.} = 1 - \frac{a \times c}{b \times l}$$

If A and L are nearly equal, we may obtain a very slow relative rotation, as in Fowler's first coiling gear, Fig. 523. Stud D supports the drum and gear, A is the fixed wheel, and a difference of about one tooth in 40 between A and L causes the latter to turn very slowly, rotating the cam E, and raising or lowering the coiling lever and guide pullies as required.

Fig. 524 has an annular wheel, but is otherwise like Case II. Opening out the train, it is found that while l's revs. are minus, those of L are plus, so

$$\text{L's revs.} = 1 + \frac{\text{A}}{\text{L}}$$

Its application is shewn to a ship's capstan; and

$$\text{Vel. Ratio} = \frac{\text{L's revs.}}{1} \times \frac{\text{lever arm}}{\text{barrel rad.}}$$

D being inserted for steadiment.

Moore's Pulley Block, Fig. 525, is a reverted train with annular wheels. Referring to the lower diagrams, the train ratio is $\frac{a \times c}{b \times l}$ and a minus rotation is induced in l or L by the relative motion of a or A.

$$\therefore \text{ L's revs.} = 1 - \frac{\text{A} \times \text{C}}{\text{B} \times \text{L}}$$

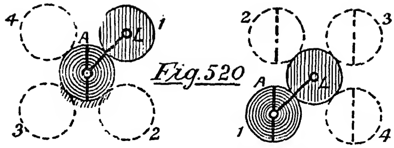


Fig. 520

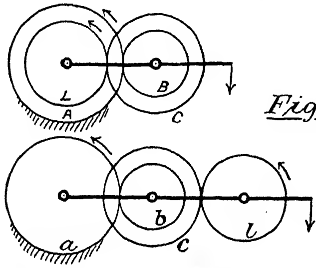


Fig. 522.

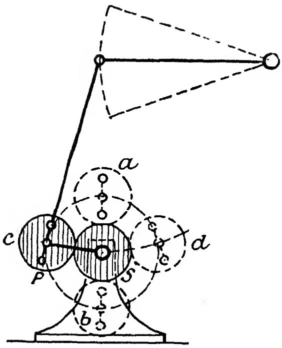


Fig. 521

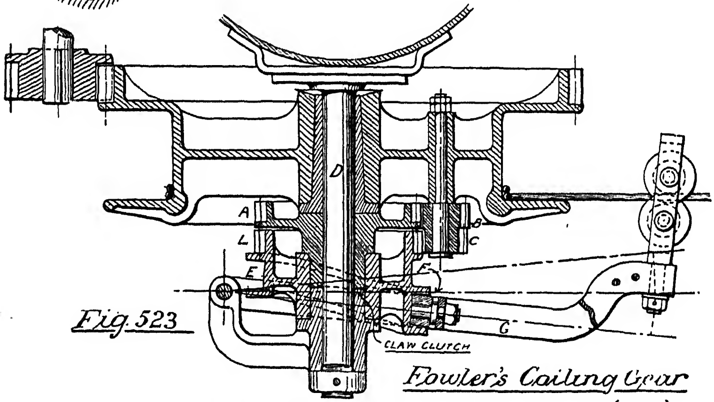
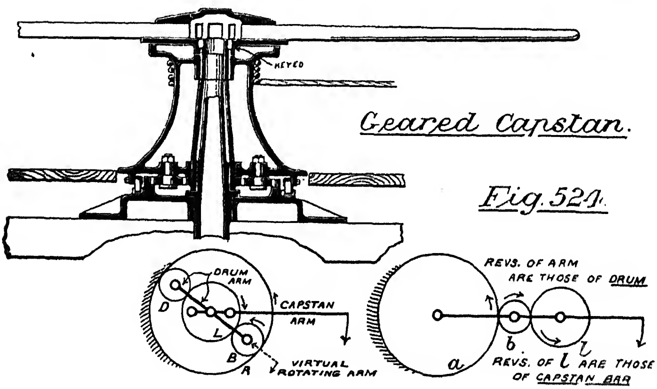


Fig. 523

Fowler's Coiling Gear

(FIRST)



Geared Capstan.

Fig. 524.

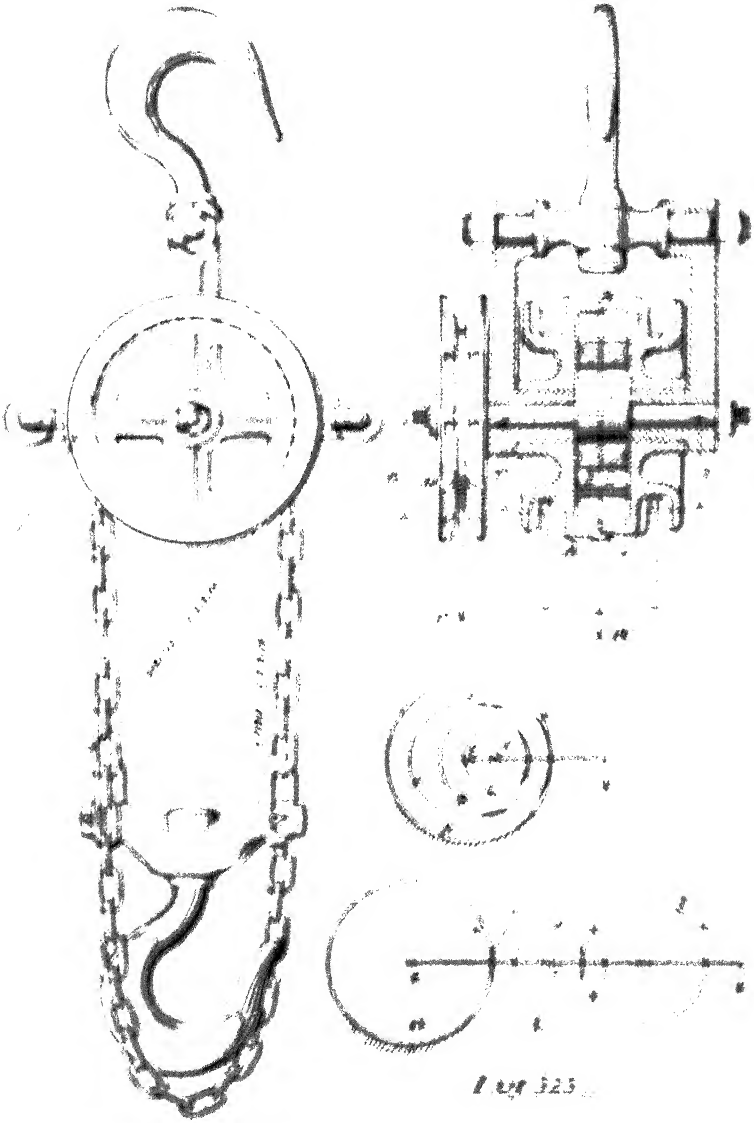


Fig. 323.

Moore's Differential Pulley Block

If A and L are nearly equal, we have a high velocity ratio. In the block, the eccentric G, corresponding to crank *ef*, is rotated by hand chain round H, so that A and L are turned oppositely, each by half their relative motion, and W's rise is due to this. Then

$$\text{P's distance} = 2\pi R$$

$$\text{W's distance} = \frac{2\pi r \times \text{L's revs.}}{2}$$

$$\text{and Vel. Ratio} = \frac{\text{P's dist.}}{\text{W's dist.}} = \frac{2R}{r \times \text{L's revs.}}$$

In the example BC has 14, A 15, and L 16 teeth. If $R = r$

$$\text{Vel. Ratio} = \frac{2}{1 - \frac{15 \times 14}{14 \times 16}} = 32 : 1$$

Another reverted train is obtained by bevel wheels, as in Fig. 526, being applied as driving gear to traction engines and tricycles. B is the arm, and A, L the first and last wheels respectively. When the front road wheel is steered ahead, A, B, and L are practically locked, and the two hind road wheels move with equal velocities; but if the front wheel be steered, say, to the left, A becomes fixed and L revolves at double speed, thus steering the engine in a much smaller curve. Fig. 527 shews a detailed section through the hind axle.

Fig. 528 is a disguised form of sun and planet motion, where L is annular and the slider-crank chain is employed. Considering A fixed, as in Fig. 520,

$$\text{L's revs.} = 1 - \frac{A}{L}$$

If A and L are nearly equal, a slow movement of L is obtained, as in Fowler's second coiling gear, Fig. 529. Eccentric B serves as crank, and D as connecting rod; A and L have the same meaning as in Fig. 528, and the cam and lever are as previously described. (*See p.* 1108.)

(6.) **Belt Gearing** has the disadvantage of slip, but is practically noiseless, and will transmit power a considerable distance (say 30 ft.) without intermediate support.

[illegible]

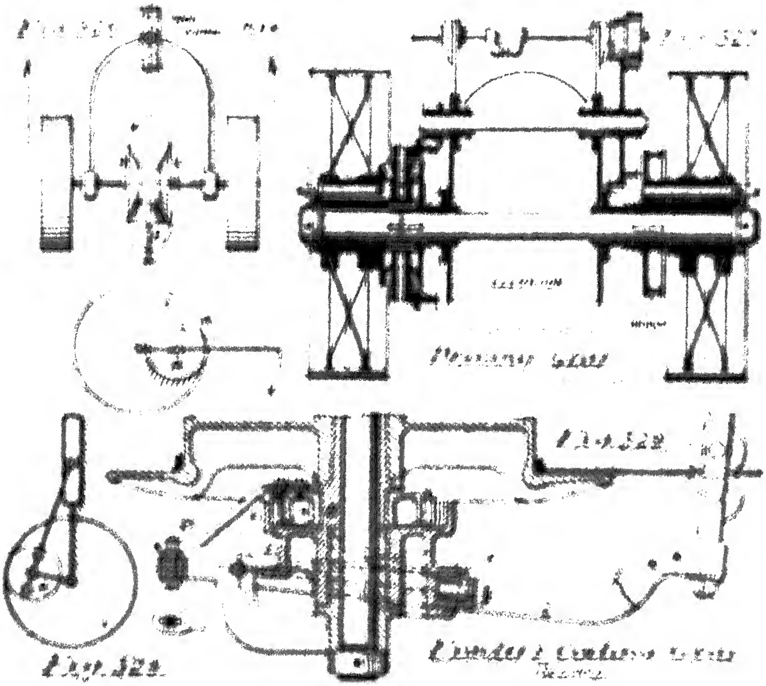


Fig. 328.

Fig. 329.

In Fig. 330 the belt embraces an angle θ, which is [illegible] circular measure. Considering a small angle α, the greatest tension without slipping being T, and the lesser tension t, these forces are balanced by reaction R, inclined to radius by the friction angle ϕ. Drawing $a\,b$ perpendicular to R forms the force diagram $A\,b\,a$, where $A\,a$ = tension T, $A\,b$ = tension t, and $a\,b$ = reaction R, and if the construction be followed through angles α, α, α, α and α, the final tension t is found for the slack side of belt. Curve $a\,d\,g$ is a logarithmic spiral, whose tangents

make a constant angle with the radii, and if the angles σ are taken small, the construction is fairly correct. But greater accuracy is secured by using the equation to the curve,

$$\frac{T_n}{t_n} = \varepsilon^{\mu\theta} \quad \text{where } \varepsilon = 2{\cdot}718\text{, the base of Napieran logarithms.}$$

More usefully the formula becomes

$$\underline{\text{Log.}\left(\frac{T_n}{t_n}\right) = {\cdot}4343\,\mu\,\frac{l}{r}}$$ (*See Appendix II.*, *p.* 867.)

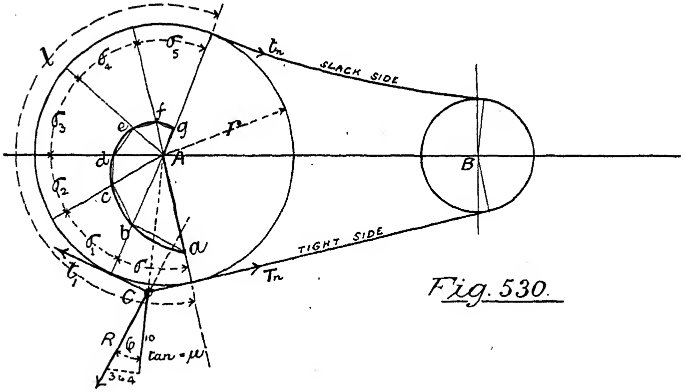


Fig. 530.

Then, the log. being known, the corresponding number is found from a table, and the formula used for any value of $\frac{l}{r}$, *even beyond* 360°.

COEFFICIENT OF FRICTION (μ) IN TENSION ELEMENTS.

Leather belting on iron pullies	·3 to ·4	1·5 if oily.
Wire rope on iron pullies	·15	not accounting for wedge action
Wire rope on leather-bottomed pullies	·25	
Hemp rope on iron pullies	·28 to ·18	

TABLE OF LOGARITHMS OF $\frac{T_n}{t}$

$\frac{T_n}{t_n}$	Log	$\frac{T_n}{t_n}$	Log	$\frac{T_n}{t_n}$	Log
$1\frac{1}{4}$	·09691	$3\frac{1}{2}$	·54407	$5\frac{3}{4}$	·75966
$1\frac{1}{2}$	·17609	$3\frac{3}{4}$	·57403	6	·77815
$1\frac{3}{4}$	·24303	4	·60206	$6\frac{1}{4}$	·79588
2	·30103	$4\frac{1}{4}$	·62840	$6\frac{1}{2}$	·81291
$2\frac{1}{4}$	·35218	$4\frac{1}{2}$	·65321	$6\frac{3}{4}$	·82930
$2\frac{1}{2}$	·39794	$4\frac{3}{4}$	·67670	7	·84509
$2\frac{3}{4}$	·43933	5	·69897	10	1·00000
3	·47712	$5\frac{1}{4}$	·72016	100	2·00000
$3\frac{1}{4}$	·51188	$5\frac{1}{2}$	·74036	300	2·47712

Driving Pull and H. P.—If two weights are slung over a pulley, as in Fig. 531, the pull on the rim of the latter will be due

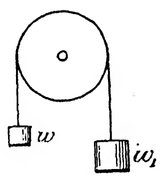


Fig. 531.

to their difference, $w_1 - w$, and as this is the same case as a driving belt,

$$\text{Driving pull} = T_n - t_n$$

$$\text{and H. P. transmitted} = \frac{(T_n - t_n)\,V}{33000}$$

$$\text{But } V = 2\pi R N \qquad \therefore \text{ H. P.} = \frac{(T_n - t_n)\,2\pi R N}{33000}$$

Strength of Belting, allowing for the joint, may be taken, so that

$$f^{\text{lbs}}\,(\text{safe}) = 320 \text{ lbs. per sq. in.}$$

and the thickness varies from $\frac{3}{16}''$ to $\frac{3}{8}''$ in single-ply belts. The width must be made sufficient to meet T_n.

Example 53.—A leather belt is to transmit 2 H.P. from a pulley 12″ diameter on a shaft making 160 revs. per m. Find (1) the tensions, when the belt embraces half the pulley rim, and $\mu = \cdot 3$: (2) the belt width when the leather is $\frac{1}{4}''$ thick.

$$(1)\ \text{Log.}\ \frac{T_n}{t_n} = \cdot 4343 \times \cdot 3 \times \frac{22}{7} = \cdot 40905 \quad \therefore \frac{T_n}{t_n} = \frac{2\frac{1}{2}}{1}$$

$$\text{H.P.} = \frac{(T_n - t_n)\, 2\pi\, R\, N}{33000} \quad \text{and}\ T_n - t_n = \frac{2 \times 33000 \times 7}{2 \times 22 \times \cdot 5 \times 160} = 131\ \text{lbs.}$$

There are two values of T_n, viz., $(t_n + 131)$ and $(2 \cdot 5\ t_n)$

$$\therefore 2 \cdot 5\ t_n = t_n + 131\ ; \quad \underline{t_n = 87 \cdot 3\ \text{lbs.}} \quad \text{and}\ \underline{T_n = 218 \cdot 3\ \text{lbs.}}$$

$$(2)\ w'' \times \cdot 25 \times 320 = T_n = 218 \cdot 3 \quad \therefore \underline{w'' = 2\tfrac{3}{4}''}\ (\textit{See p. } 1110.)$$

Tension in Belt due to Centrifugal Force may be examined similarly to the fly wheel at Fig 353. The weight of a cubic inch of leather (w) is $\cdot 0358$ lbs., and the stress per square inch becomes

$$\frac{12\ wv^2}{g}\ \text{lbs.}$$

$$\therefore \text{Total tension on tight side} = T_n + \frac{\cdot 43\ w\, t''\, v^2}{g}$$

which is the total force the belt must resist at high speeds.

Creep, Slip, and Speed.—As the belt tension changes from T_n to t_n, a small retrograde movement or creep occurs due to release of tension, causing the follower to revolve at a slightly decreased rate. The result is known as slip, and represents a loss of about 2 per cent. The speed of belting should not exceed 3000 to 4000 feet per m.

Length of Belt (Figs. 532 and 533).—The length between centres c should not be less than 6 times D if much power is transmitted, though much less is used with light pressures. It may be as much as 30 feet. Horizontal belts give better results than vertical ones, and some inclination should always be given upright belts if possible. Taking the *open belt*, Fig. 532,

$$l_1 = \sqrt{\left(\frac{D-d}{2}\right)^2 + c^2} \quad ; \quad l_2 = \frac{\pi D}{2} \quad ; \quad l_3 = \frac{\pi d}{2}$$

and $\underline{\text{Total length of belt} = 2l_1 + l_2 + l_3}$

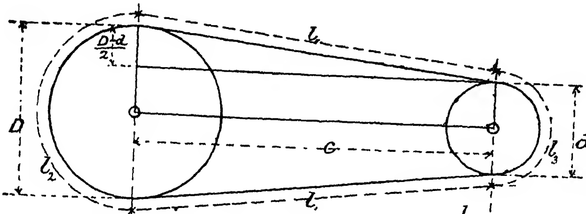


Fig. 532.

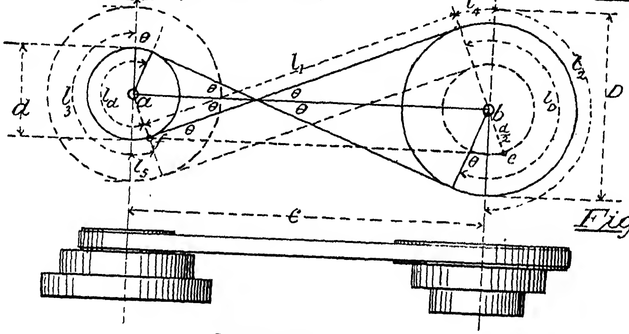


Fig. 533.

Length of Belts

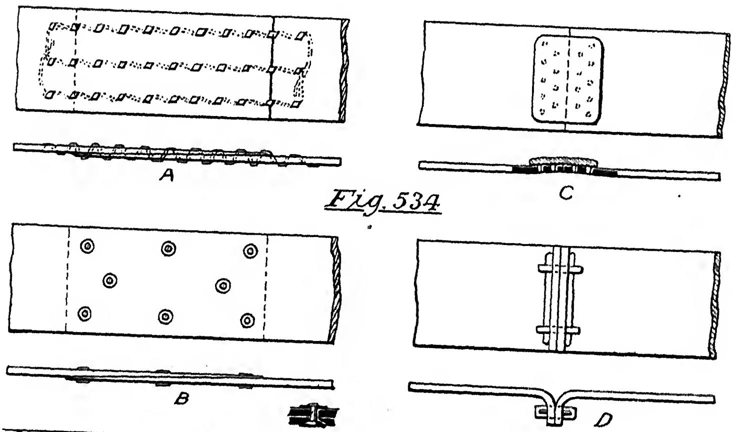


Fig. 534

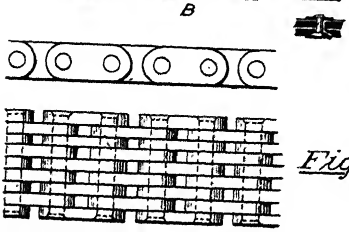


Fig. 535.

Details of Belting.

In the crossed belt [illegible] and [illegible]. The [illegible] angles θ are [illegible]

$$[illegible]$$

Then [illegible]

$$[illegible]$$

and Total length of belt = [illegible]

If $R + r$ be constant throughout the pairs of cones, a crossed belt will fit equally well on any part. Thus in Fig. [illegible], when the belt is changed to the dotted position, [illegible] has a constant value, and as circumferences vary as radii, the sum of the circumferences will also be constant. This is not exactly true for an open belt, but may be safely reckoned in ordinary practice. The diameters should always be measured to centre of belt thickness.

Belt Fastenings. The common method of connecting the ends of leather belts is by lacing, as [illegible] Fig. 634. There are, however, many patented metal fastenings, as Harris's, at c, being a spiked plate having the points burred over after connection; and Jagger's, at d, where the belt is first bent over. The belt being laid on the pulleys and tightly stretched, has the length marked, and the joint made while lying round the shafts; the belt is then placed upon one pulley, and gradually drawn on to the other by lying it on the rim and slowly rotating the latter. Large belts must be stretched by means of clamps. Tullis's chain belting, Fig. 635, has a [illegible] grip by using the edge of the leather, but the method is expensive.

Advancing and Retreating Sides.—Let A, Fig. 636, be a pulley whose belt enters at B and leaves at C. It will remain on the pulleys while exactly at right angles to the shaft, as at B. If, however, the advancing side be deviated as at D, the belt will slip off to the left, but the retreating side may be deflected, as at E, without any harm. In a *curved pulley*, Fig. 637, the belt will ride on the large diameter with safety, for if placed at F, the pull causes a deviation which moves the belt to the right, as at G the

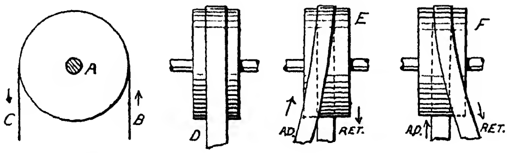


Fig. 536.

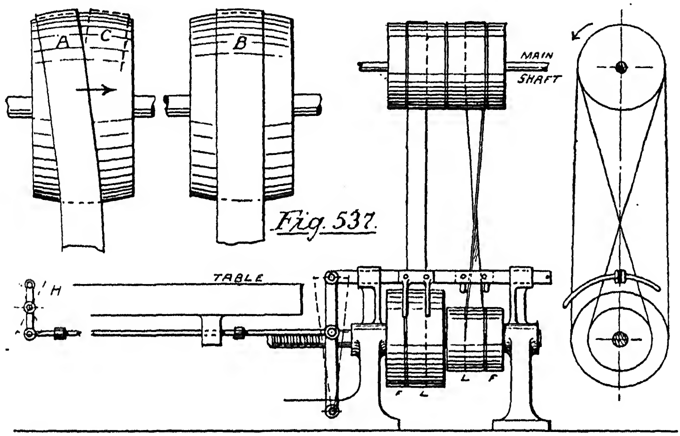


Fig. 537.

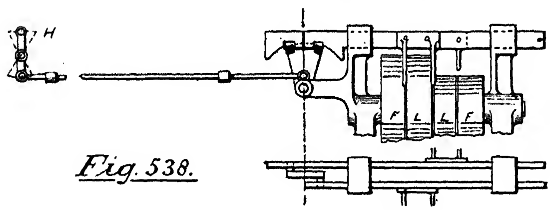


Fig. 538.

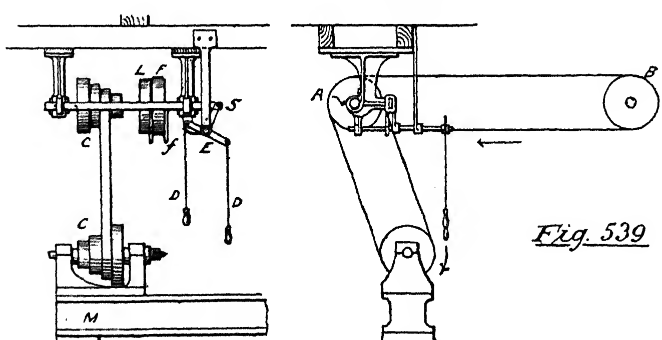


Fig. 539

Belt Driving.

movement is leftward, and the final position is that at B. The radius of curvature should be three to five times the pulley width.

Countershafting and Speed Cones.—Fig. 539 shews how a shop machine M may be driven so as to be started and stopped without affecting the main shaft revolutions or removing the speed cone belt C C. B is the main shaft and A the countershaft, the latter having fast and loose pullies L and F. The fork *f* on the striking bar S then grasps the advancing side of the belt, and is moved to right or left by pulling the handles D, which act on the belt crank L.

Quick return is obtained by the belting at Fig. 538. An open strap turns the advancing, and a crossed strap the returning pulley, and in each case there is a narrow fast pulley and a broad loose pulley. The fork is shifted automatically at either end of stroke, and the machine stopped by placing both belts in position shewn, from the handle H. The total width of pullies may be reduced to four times belt width by the arrangement shewn below, where two striking bars are employed with which the black tappets only engage at certain times. Many belt examples will be found in Part I.

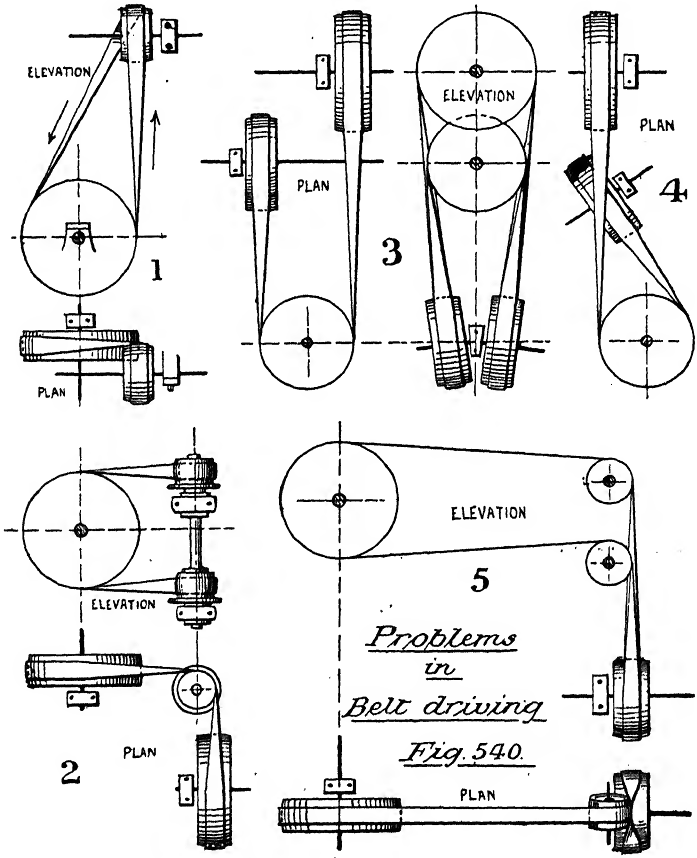
√ **Problems in Belt Driving.**—The more difficult cases are solved in Fig. 540, and will be understood if it be remembered that the advancing side of the belt must lie at right angles to the shaft, while the retreating side may make any deviation.

Pullies for Belt Driving are usually split, for convenience in fixing. Fig. 541 shews the construction of a cast iron, and Fig. 542 of a wrought iron pulley. The former should have curved arms if more than 12″ diameter (see p. 67), and the latter is adopted for lightness with high speeds or large pullies. Fig. 543 shews a section through a pair of fast and loose countershaft pullies, which need not be split.

(7.) **Cotton-Rope Gearing** is much in favour for spinning and weaving mills, and has been successfully applied to travelling cranes and dynamo driving. For mills, the flywheel rim has the section shewn in Fig. 544, and the ropes lie in wedge grooves.

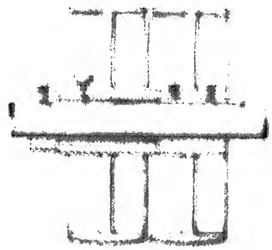
With a flat pulley the resistance to slip would be $P\mu$, but in the grooved pulley shewn the resistance is $2\ R\mu$, there being two

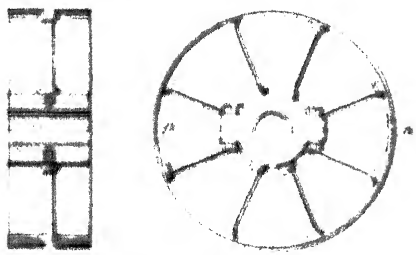
friction surfaces. The grip is greater in the second than the first case in the ratio $\frac{2R}{P}$: 1. From the force diagram, $\frac{2R}{P}$ = cosec. $22\frac{1}{2}°$ = 2·6131, and $\mu\frac{2R}{P}$ = ·28 × 2·6131 = ·732, which should be substituted for μ in the tension formula already given.

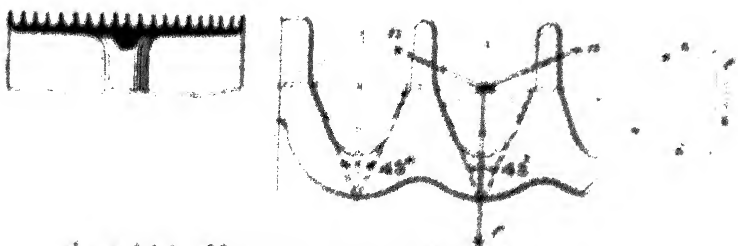


Problems in Belt driving
Fig. 540.

Messrs. Jno. Musgrave and Sons, of Bolton, have fitted up a large number of mills with cotton-rope driving, and the following remarks and tables are the result of their experience as given in







Pulley for Cotton Ropes.

bending, the life of a cotton rope may be twelve years in good hands, and has even reached seventeen years while still in good order. Taking the area of circumscribing circle, the breaking stress is 4 tons per sq. in., but a factor of 45 being adopted, 200 lbs. per sq. in. is the safe load, or about the same as leather belting.

DATA FOR COTTON ROPES, WHEN V = 4700 FT. PER M.
(Messrs. MUSGRAVE.)

Dia. of rope.	Area of circle.	Weight per foot.	$(T_n - t_n) + \frac{12wv^2}{g}$	Centrifugal stress $\frac{12wv^2}{g}$	Effective tension $(T_n - t_n)$	H. P. transmitted.	Centres pulley grooves.	Dia. of smallest pulley.
ins.	sq. ins.	lbs.	lbs.	lbs.	lbs.		ins.	ins.
$\frac{1}{2}$	·1963	·081	47	16	31	4·43	$\frac{7}{8}$	15
$\frac{5}{8}$	·3067	·125	72	24	48	6·84	1	18
$\frac{3}{4}$	·4417	·184	106	35	71	10·07	$1\frac{1}{8}$	22
$\frac{7}{8}$	·6013	·25	144	48	96	13·67	$1\frac{5}{16}$	26
1	·7854	·33	190	63	127	18·05	$1\frac{1}{2}$	30
$1\frac{1}{4}$	1·2272	·51	294	98	196	27·9	$1\frac{13}{16}$	37
$1\frac{1}{2}$	1·7671	·74	426	142	284	40·48	$2\frac{1}{8}$	45
$1\frac{3}{4}$	2·4053	1·00	576	192	384	54·7	$2\frac{1}{2}$	52
2	3·1416	1·30	750	250	500	71·10	$2\frac{3}{4}$	60

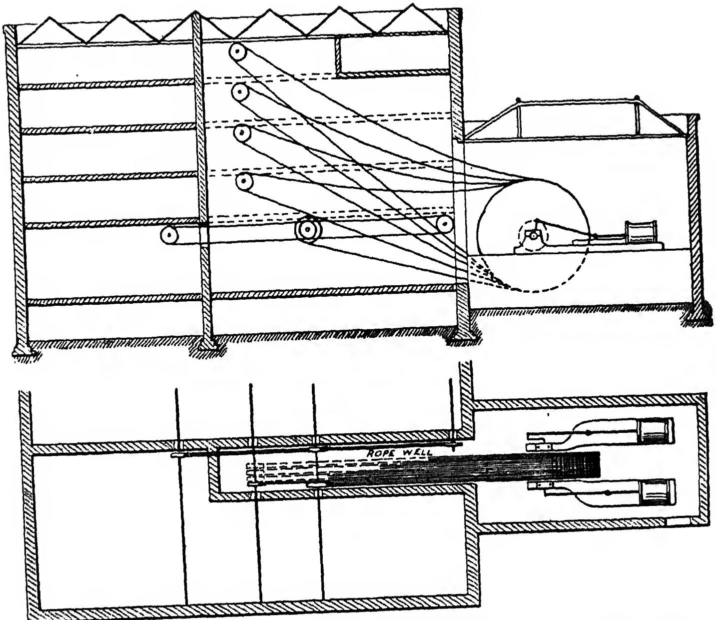
The centrifugal stress is *weight per foot* × $v^2 \div g$, and the fourth and sixth columns assume that $t_n = \cdot 2\, T_n$ which gives $T_n : t_n :: 5 : 1$. From the tension formula,

$$\text{Log. } 5 \text{ or } 69897 = \cdot 4343 \times (\mu \times 2\cdot 613) \times \frac{\cdot 55 \times 2 \times 22}{7}$$

$$\therefore \mu = \frac{4\cdot 893}{27\cdot 463} = \cdot 18$$

$1\frac{3}{4}''$ is the usual diameter for main rope.

Fig. 545 shews a spinning-mill driven by cotton ropes, the power being given to five floors by separate sets of ropes, a good arrangement in case of breakdown. The slack side being uppermost gives a large arc of contact. Fig. 546 shews a travelling crane. The rope is endless, passing round the pullies D, H, A, G, F, and E in succession, and kept taut by the weight at H. Worm gear is used in taking off the power, at E for travelling, at B for lifting and lowering, and at C for cross-traversing; and either rope is put in gear by the press pullies *a*, *b*, actuated by hand levers P, Q E is reversed by friction gear worked from handle L.



Mill driven by Cotton Ropes. *Fig. 545.*

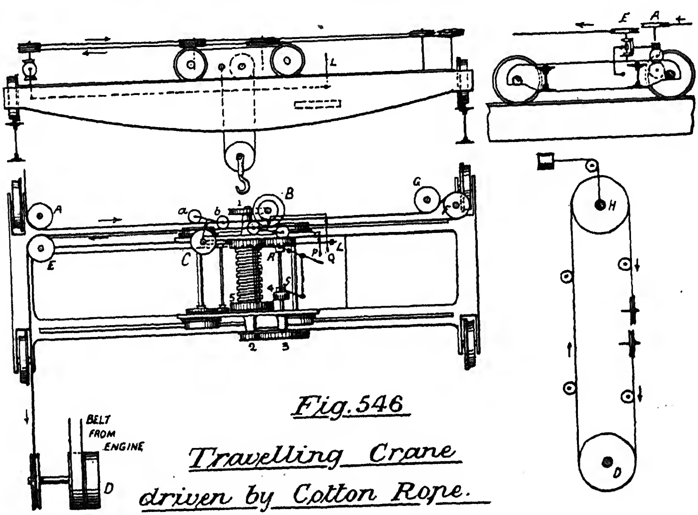


Fig. 546

Travelling Crane driven by Cotton Rope.

It should be noted that horse-power, depending upon *pressure* × *speed*, may be obtained either by a large value of the one or other quantity. Thus cotton-rope driving depends upon a low pressure and high speed, but high-pressure driving will now be considered.

(8.) **Wire-Rope Gearing**, introduced by Hirn in 1851, and called by him 'telo-dynamic transmission,' has since been used in many long-distance cases, for example :

1. From turbines to distant mills.
2. For steam ploughing.
3. In collieries: both for hauling and raising.
4. For travelling cranes.
5. Funicular railways and cable tramways.
6. Boat towing on canals.

The rope is of steel wire, with hemp or steel core, and six strands of from 7 to 12 wires each. The wear is more uniform if the strands twist in the same direction as the rope, as in Lang's patent. The following table refers to the latter ropes :

Circumf. of rope.	Dia. of circle.	Breaking Stress.		Dia. of smallest sheave.	Construction.	
		Hemp Core.	Steel Core.		No. of strands.	Wires in each strand.
ins.	ins.	tons.	tons.	ins.		
4	$1\frac{1}{4}$	34	51	24	6	12
$3\frac{1}{2}$	$1\frac{1}{8}$	27	40	21	6	12
3	$\frac{15}{16}$	19	28	18	6	12
$2\frac{1}{2}$	$\frac{13}{16}$	14	21	12	6	12
$2\frac{1}{4}$	$\frac{3}{4}$	10	15	10	6	12
2	$\frac{5}{8}$	8	12	10	6	12
$1\frac{3}{4}$	$\frac{9}{16}$	6	9	8	6	10
$1\frac{1}{2}$	$\frac{1}{2}$	$4\frac{1}{2}$	$6\frac{1}{2}$	6	6	8

The wire core does not affect the safety of the rope in bending round pullies.

Pullies for Wire Rope.—The section is shewn in Fig. 547, having a groove filled with leather on edge which is afterwards turned: μ then is ·25. Fowler's clip pulley, Fig. 548, has its rim divided into a series of toggles, the mere pull of the rope causing great grip, as shewn at A. E is a huge screw on the pulley rim which permits adjustment, after which the bolts are re-inserted. The clip pulley has enabled wire rope to be applied in many cases hitherto unsolved. Fig. 549 shews a guide pulley.

At Fig. 550 a turbine (or horizontal water wheel) T_b drives a distant workshop. A B is termed a relay, which should not exceed 500 feet, and C C are guide pullies. Fig. 551 shews two methods of steam ploughing: (I) is the 'direct' system, engaging two engines which wind up the rope alternately, and advance along the headland between bouts; (II) is the 'roundabout' system, where a portable engine A drives a windlass B in either direction as required. C, D, are self-acting anchors, which resist the pull of the rope; and as the slack-rope anchor automatically winds itself in the direction of the claw anchor F, the tight-rope anchor is meanwhile fixed. G is a rope porter. Fig. 552 serves to explain underground haulage. An endless rope is used at (I), being crossed at J to obtain a greater grip on the clip pulley H, and tightened at E with a heavy weight. (II) employs a pair of winding drums C, as in the case of steam ploughing. The haulier attaches his wagon by scissors grip at A. The up and down rails are omitted for clearness.

Fig. 553 represents the lifting gear at a pit-head. The cages move in opposite directions, and while one drum is winding the other pays out, a brake being attached to each. When the mine is very deep, the conical drum, Fig. 553*a*, is advisably employed. It is on the fusee principle. When the cage is near the bottom the load is greatest, due to rope weight, and the drum radius is decreased, so that an approximately even turning moment is required throughout the lift. Overwinding has constituted a serious danger, and may be avoided either by automatic reversing gear on the engine, or the detaching hook in Fig. 556 (Walker's). The mouth of the hook is usually closed by the ring A, but if the engine be over-run the hook attempts to pass through the ring B, in the beam C above the

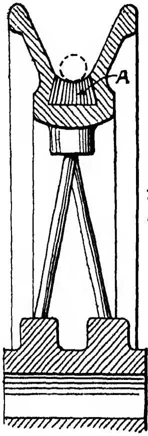


Fig. 547

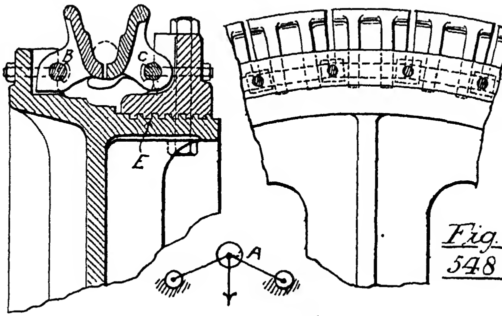
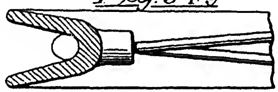


Fig. 548

Fig. 549



Pullies for Wire Rope.

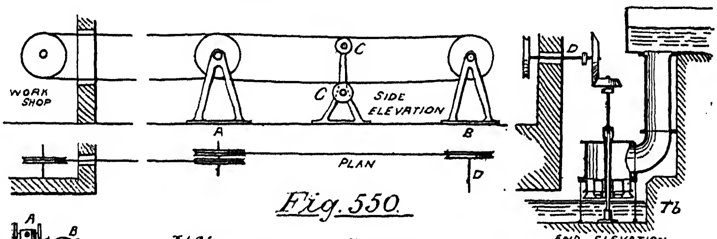
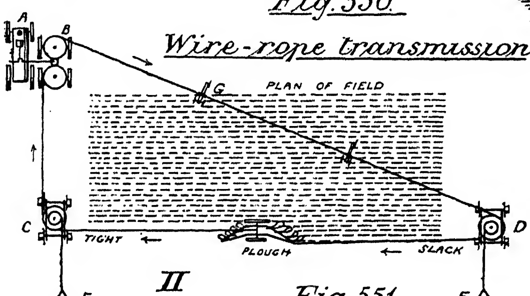


Fig. 550.

Wire-rope transmission



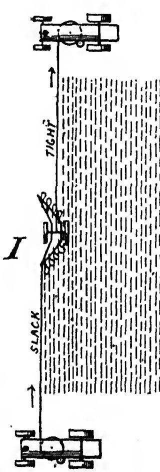


Fig. 551

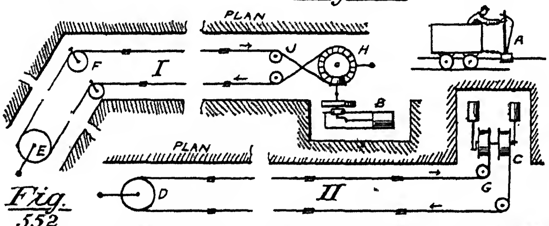


Fig. 552

pit-head; and A is thereby caught, being slipped relatively downward. The jaws then open or catch on B, as at D.

Fig. 554 shews Fowler's travelling crane driven by wire rope round clip pullies. A is the rope arrangement, and the power is distributed for travelling at C C, cross traversing at D, and lifting at E F. The last is accomplished by the rotation of screw F, which shortens the lifting chain attached to nut E. The arrangement is suitable for very heavy cranes.

Cable tramways are useful for bad inclines. An endless rope travels in a conduit A, Fig. 555, and the car carries the gripping lever B, which, when moved to the vertical, raises the rollers C C, and brings the jaws D D together. Some jerk is, of course, unavoidable.

Fig. 557 is a towing arrangement adopted on some German canals. A rope is anchored on the canal bottom, and the tug winds itself along by the engine-driven clip pulley. The rope serves as a rail, and with the pulley forms a kinematic pair.

In wire-rope transmission the tension ratio is usually 2 : 1 and the speed 3000 to 6000 feet per m. The stresses in the rope are due to:

(1) Weight of rope and the form of hanging curve.
(2) Bending of rope round pulley.
(3) Centrifugal force.

(1) In Fig. 558 the catenaries may be considered as parabolas for all practical purposes. Then the tangent T A being drawn, by bisecting C D at A, the force diagram will give the value of T, in terms of W the weight of rope between the pullies, and B the pressure on the bearing. The weight of wire rope per foot = $(1{\cdot}34 \times d^2)$ lbs. (2) Taking the general bending formulæ,

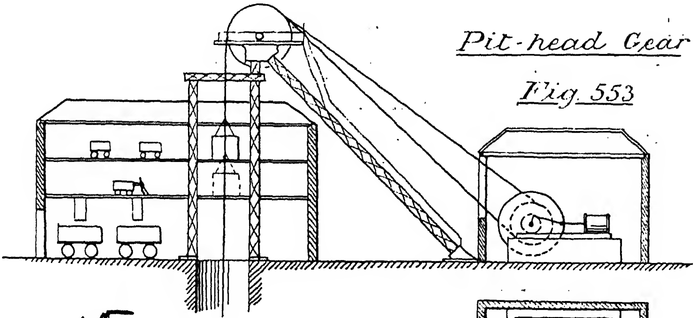
$$B_m = \frac{E\,I}{\rho} = f\frac{I}{y} = fZ \qquad \text{and } I = Zy$$

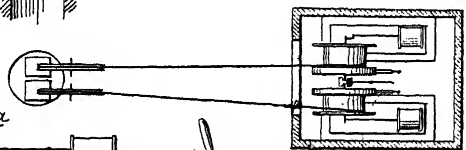
where ρ = radius of pulley, and y that of the rope-wire:

$$\frac{E\,Zy}{\rho} = fZ \quad \text{or } \underline{f^{\text{lbs}}} = \frac{Ey}{\rho} = 30{,}000{,}000\ \frac{y}{\rho}.$$

Pit-head Gear

Fig. 553





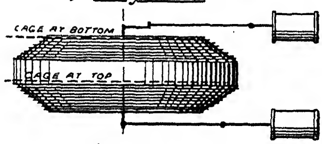


Fig. 553a

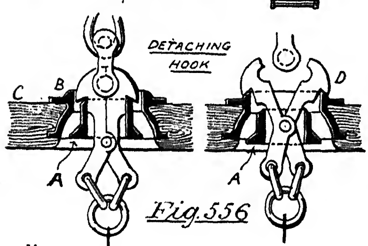


Fig. 556

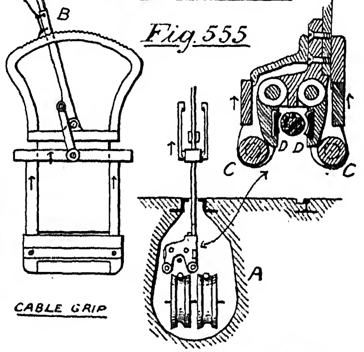


Fig. 555

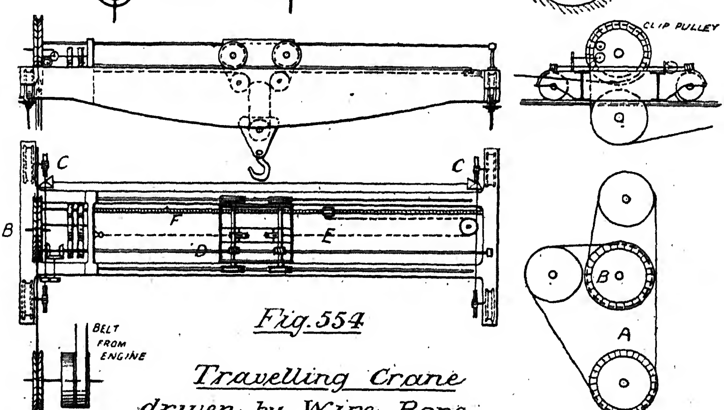


Fig. 554

Travelling Crane driven by Wire Rope

(3) has been already treated for belt and cotton rope. The safe strength of the rope must meet the combined stress (1) + (2) + (3), but the driving tensions T_n and t_n caused by T will both be decreased by the stresses (2) and (3).

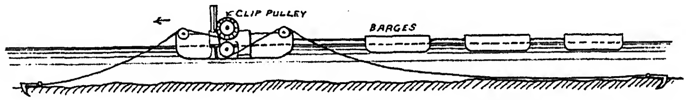


Fig. 557. TOWING

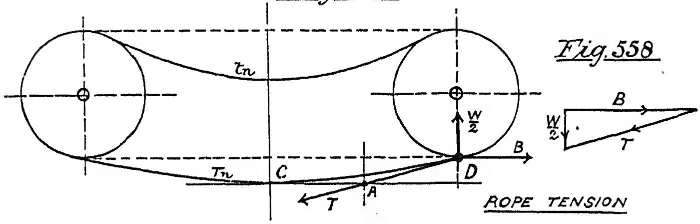


Fig. 558 ROPE TENSION

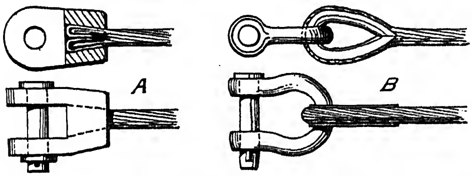


Fig. 559

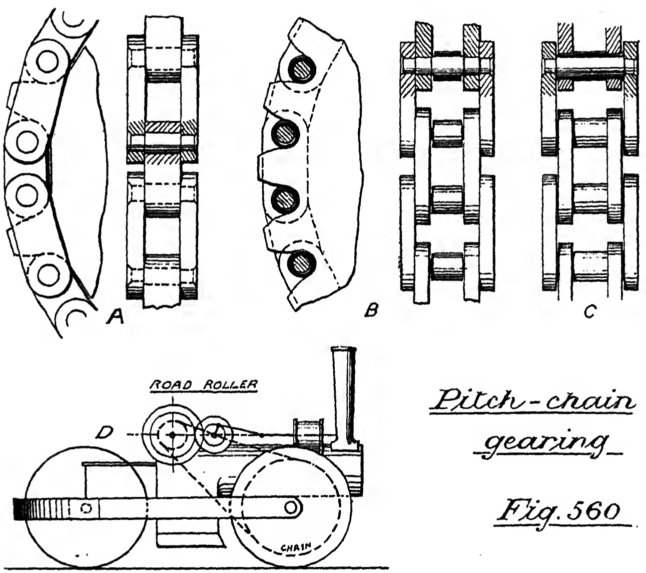
Shackles for Wire Rope

Two shackles are shewn at Fig. 559. At A the wires are bent back and soldered, giving a joint equal to the rope strength; but B is wrapped round a wrought-iron eye and then spliced, the joint having but 50 or 60 °/ₒ of the rope strength.

(9.) **Pitch-chain Gearing** serves the purpose of belting where positive driving is required or considerable pressures are to be transmitted. If high speeds are employed, the gear should be exceptionally well made. Much power is lost in friction, and the journals must be adjustable to take up stretch or wear.

Fig. 560 shews three forms of chain. At A the teeth bear on solid inner links, but at B and C they engage with the pins, and the smaller pitch obtained gives more regular driving. There are

two sets of friction surfaces; the teeth on the pin and the pin on the inner links. B decreases the friction of the former surfaces by the introduction of rollers, and at C the latter surfaces are enlarged by riveting a ferrule to the inner links. The pins should be shouldered, so that the links may work clear at the sides; and the teeth are involute curves having the arc of pin centres as base circle.



Pitch-chain gearing

Fig. 560

D is a road roller supplied with pitch chain, and Fig. 569 an electric car driven by chain from a dynamo. Cycle driving is a well-known application. (*See p.* 1111.)

(11.) **Compressed Air** is of great advantage as a long-distance transmitter, and as such has been used for motors in mines, for tunnelling machines, and for distribution from central stations in towns. In mines and workshops, the exhaust serves also for ventilation. To be effective the compressors should be on a somewhat large scale, and the arrangement is shewn in Fig. 561, where steam in cylinders A A is used to compress air in the

cylinders B and C, which is conveyed thence to a storage receiver for distribution by main to the various motors.

When work is done on a gas, the temperature is raised, by reason of the conversion of that work into heat; and again, when a gas does work its temperature is lowered, for a reverse reason.

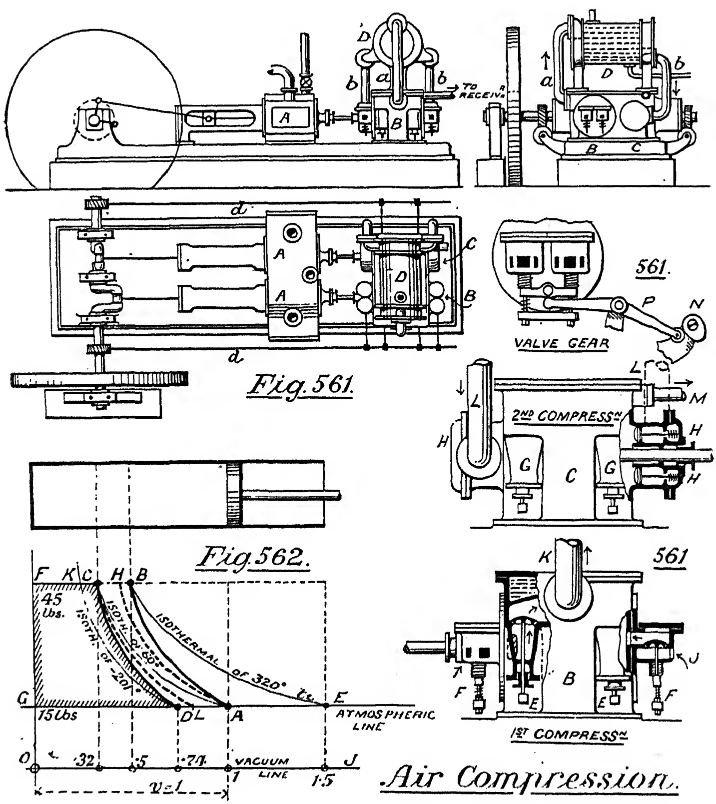


Fig. 561.

Fig. 562.

Air Compression.

The changes are shewn in Fig. 562. Draw co-ordinates O F for pressure and O J for volumes; then let the piston commence with a cylinder volume 1, opposite A, and atmospheric pressure 15 lbs. at 1 A, the temperature being 60° F. A C is a hyperbola or *isothermal* ('at constant temperature') of 60°, while E B and D K

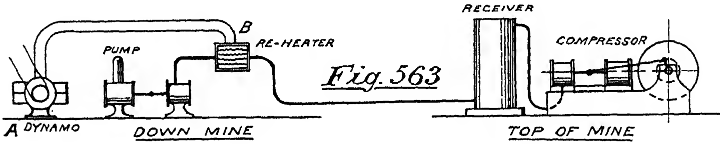
are parallel hyperbolas at 320° and −201° respectively. In compressing the air without subtracting heat, its temperature rises to 320°, and the pressure curve is the *adiabatic* ('no heat passing through') from A to B, the volume being now reduced to ·5 with pressure 45 lbs. Suppose the temperature next to lower to 60° during transit to motor, pressure remaining constant, which is practically true, the volume will decrease from B to C, viz., to ·32. Next let the air expand behind the motor piston, without adding heat, and its pressure will fall to 15 lbs., while its volume becomes ·74, and the expansion curve will be the adiabatic C D, the final temperature of which is − 201°. The area A B F G shews the work given to the gas, C D G F that restored to the motor, and the loss due to cooling is the area A B C D, being here about 27°/₀. In practice the curves would more nearly approach the thick dotted lines, but there are losses in steam cylinder, main, and motor, which may reduce the efficiency to 30°/₀ instead of the 73°/₀ shewn. (*See also pp.* 773 *and* 881.)

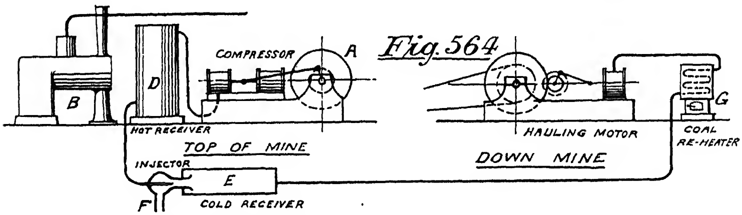
Formerly simple steam-engines were employed as compressors, but these are now replaced by compound engines; much improvement too, has been made in the methods of cooling. It being granted that isothermal compression is the ideal condition, the old water-jacketing proved inefficient, as removing the heat after adiabatic compression had been permitted. Water pistons were little better, being cumbrous and slow; while water spraying in the air cylinders both spoilt the cylinders and gave but a slight further advantage, for the time was too short for the heat to be taken up. The greatest improvement was made by the introduction of two-stage compression, or the performing of the work in two cylinders, with an intermediate cooler. It can easily be shewn that if a succession of such cylinders and intermediate coolers be used, the compression may be truly isothermal, thus gaining a large portion, but not all, of the lost work area, for loss in the motor may still occur. The blocking of the ports with ice or snow on account of the low temperature of the motor exhaust caused trouble which an attempt to overcome was made in 1887, by re-heating the air entering the motor, but the economic results proved such a surprise, that the method has ever since been followed, with increasing success; and it is now clearly under-

stood that the employment of fuel in a re-heater is attended with some six times the economy of the same fuel used in a steam boiler furnace. Figs. 563–4 shew at B an electrical re-heater formed of resistance coils in the circuit of the dynamo A, and at G a stove re-heater through which the air pipe passes. Still another saving is obtained by air injection. As heat is nothing but a form of work, it may be made to do work as soon as generated, instead of being allowed to dissipate. In Fig. 564 this is done by allowing the hot air to pass from the receiver D through the injector nozzle F, and thus an additional quantity of air is drawn into the cold receiver E to fill up the loss caused by shrinkage during cooling. The air being compressed to 100 lbs. at a temperature of 484°, is reduced to 50 lbs. in E, with a temperature of 201°; but the gain is certain, for the heat has been made to do work.

Much mechanical improvement has been introduced in the compressors, such as the use of lever-lifted valves instead of air-moved flaps, avoiding wire-drawing. Clearance spaces have been much reduced, and the mains increased so as to bring the air velocity below 30 ft. per sec. Referring now to Fig. 561, the cylinders A A are compound high and low pressure, and the air enters first the suction valves F F of the cylinder B. Leaving by the valves E E, it passes by *b* to the surface condenser D, and then to the second cylinder C, which it enters by H H and leaves by G G. Finally it passes by pipe M to the storage receiver. The valves are lifted by levers P, moved by cams N on shafts *d d*.

The Paris Compressed Air Company delivers about 8000 H.P. from two central stations, through thirty-five miles of piping, the further motor being 4½ miles away. Prof. Kennedy measured the efficiencies in 1889, and found that for one I.H.P. in central engine, the customer received ·39 I.H.P. with cold air and ·47 with air re-heated just before entering his motor. With two-stage compression and other improvements, François shewed in 1891 that a total efficiency of ·46 could be reached with cold air, ·65 with hot air, and ·8 if the hot air was sprayed with water; which results have since been approached. This assumes efficiencies of compressor, main, and motor at ·9, ·96, and ·93 respectively.





Compressed Air Transmission.

(12.) **Hydraulic Transmission.**—Refer to Chapters VII. and XI.

(13.) **Electric Transmission** can only be briefly described. Faraday, in 1831, discovered magnetic induction, by which a current is generated in a closed circuit wound on a bobbin, when the latter is moved before the poles of a permanent magnet (A, Fig. 565.) Pixii, Clarke, and others thereupon, in 1832, devised the magneto electric battery B, where the bobbins are rotated, and introduced the commutator to reverse the alternate currents formed at C and thus 'straighten out' the total current. Nollet, Van Malderen, and De Meritens improved this machine up to the year 1871, dispensing with commutator, and thus producing alternating currents (D). Dr. Siemens devised the **H** armature E in 1857, working with compound magnet, and in 1866 Wilde employed a small Siemens machine F with commutator to excite the electro-magnets G of a much larger machine, and thus avoid the necessity for large permanent magnets. The progress now was very rapid, and in 1867 Siemens, Wheatstone, and Varley separately discovered the 'dynamo-electric principle,' by which the machine was made wholly self-exciting, the mere residual magnetism in the soft iron core, whether new or after use, being sufficient to commence the current, which then gradually increased up to its maximum. K is a Siemens dynamo with **H** armature and commutator, the currents being thereby

continuous. In [illegible] armature [illegible] which ends [illegible] ciple, and [illegible]

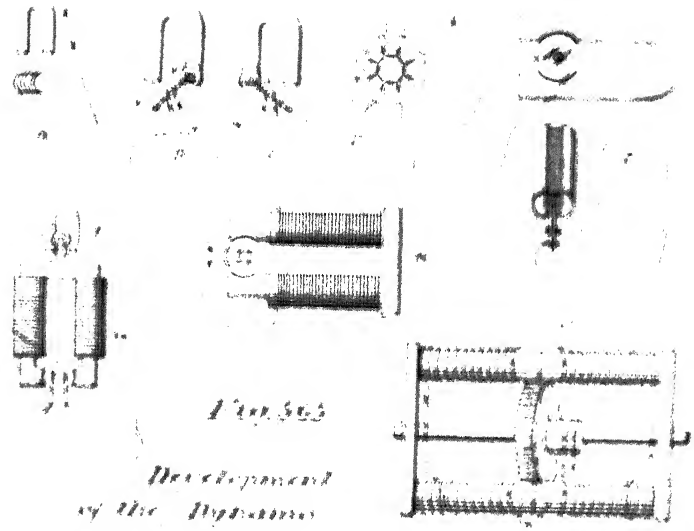


Fig. 565

Development of the Dynamo

in *Appendix II*. It is most important to notice, however, that all dynamos are *reversible*, that is, may be used either as generators or motors.

Fig. 566 shows a generator dynamo at A and a motor dynamo at B. Power given to A will be transmitted electrically to B, and the latter will rotate, thus returning the mechanical energy deposited at A; but the rotation of B will be in the reverse direction of its rotation. This is called *direct transmission* (See § [illegible]).

Electric energy may, however, be stored. In [illegible] it was found that when the electric circuit was completed by dilute acid between platinum poles, oxygen was given off from the positive and hydrogen from the negative wire (a, Fig. 567). Ritter, in 1803, showed that the platinum poles and liquid constituted a battery, which would *return the current*. In 1859 Planté made this fact of use by building the storage battery B, which consisted

of two sheets of lead *a* and *b*, rolled into a spiral, with insulating strips between, and placed in a vessel containing dilute sulphuric acid. Charging till the positive surfaces were coated with lead dioxide and the negative with metallic lead, the plates were in such a chemical condition as to constitute a return battery. Faure shortened the time of charging by coating the plates with red lead (the lower oxide), and covering this with parchment tied with strips. The only difference in action was that spongy lead

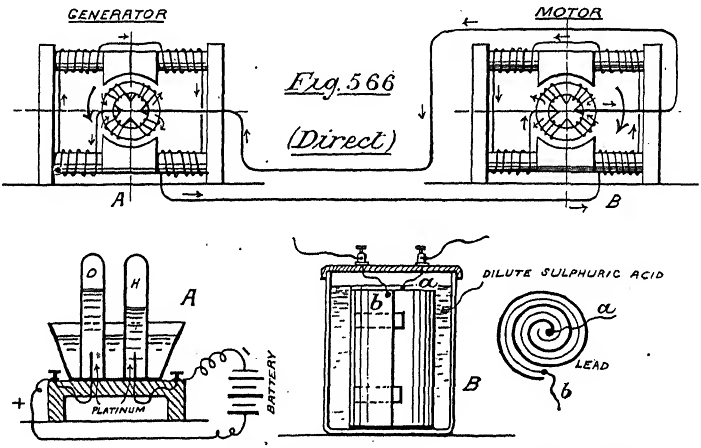


Fig. 566 (Direct)

(By Storage) Fig. 567.

Electric Transmission.

was formed at the negative plate, thus giving a large surface. Present storage or secondary batteries (otherwise accumulators) are on Planté or Faure's principle, and do not really store electricity, but change electrical energy into that of chemical separation. They are useful where the demand for power is intermittent, and are fairly effective, the leakage during a few days being but small. (*See pp.* 958 *and* 1118.)

Efficiency.—The work lost during transformation in a dynamo may be as low as 8 %, though it more often reaches 15 % or 20 %. A greater loss usually occurs, however, between generator

and motor, the resistance of the circuit causing dissipation of energy as heat. If

C = current in ampères,
E = electromotive force in volts,
R = resistance in Ohms,
Q = quantity in Coulombs,
W = work in foot pds.,

Then:

$$\text{H. P} = \frac{C^2 R}{746} = \frac{E\,C}{746} = \frac{E^2}{746\,R}$$

$$W = \cdot 737\,E\,Q$$

$$C = \frac{E}{R} = \sqrt{\frac{746\text{ H. P.}}{R}} = \frac{746\text{ H. P.}}{E}$$

$$E = CR = \frac{746\text{ H. P.}}{C} = \sqrt{\text{H. P. } 746\,R} = \frac{W}{\cdot 737\,Q}$$

$$R = \frac{E}{C} = \frac{746\text{ H. P.}}{C^2} = \frac{E^2}{746\text{ H. P.}}$$

$$Q = \frac{W}{\cdot 737\,E}$$

If l be the length of a circuit in feet, both lead and return, and a = sectional area of wire in sq. in.,

$$R \text{ at } 60^\circ \text{ F} = \frac{l}{a} \times \frac{8 \cdot 4}{1{,}000{,}000}$$

when copper wire is used. Also if the E. M. F. drops from E to e and the current from C to c in flowing from generator to motor, and if W is the work put in by generator and w that received by motor,

$$\text{Efficiency of circuit} = \frac{w}{W} = \frac{e}{E} = \frac{C - c}{C}$$

Example 54.—A dynamo driven by turbine can generate 50 ampères at 300 volts. The current is carried by two No. 6 W. G. copper wires to drive a workshop motor $\frac{1}{4}$ mile away. Assuming the commercial efficiency of the generator as 86%, and that of the motor as 84°/₀, find the mechanical efficiency of the whole system.

$$\text{H. P. given out by generator} = \frac{EC}{746} = \frac{300 \times 50}{746} = 20$$

$$\text{H. P. Turbine must give to dynamo} = \frac{20}{\cdot 86} = 23{\cdot}25$$

$$\left.\begin{array}{l}\text{R of circuit, taking lead and return} \\ \text{(dia. of wire} = \cdot 192\text{, area} = \cdot 03\text{,} \\ l = 2640 \text{ ft.)}\end{array}\right\} = \frac{2640 \times 8{\cdot}4}{\cdot 03 \times 1{,}000{,}000} = \cdot 74 \text{ ohms}$$

$$\therefore \text{ H.P. lost in wire} = \frac{C^2R}{746} = \frac{50 \times 50 \times \cdot 74}{746} = 2{\cdot}48$$

$$\left.\begin{array}{l}\text{H. P. delivered to motor} \\ \text{is that generated less} \\ \text{that lost in wire.}\end{array}\right\} = 20 - 2{\cdot}48 = 17{\cdot}52$$

$$\text{H. P. available at shop shafting} = 17{\cdot}52 \times \cdot 84 = 14{\cdot}71$$

But H.P. given by turbine was 23·25

$$\therefore \underline{\text{ Gross efficiency}} = \frac{\text{H. P. taken out}}{\text{H. P. put in}} = \frac{14{\cdot}71}{23{\cdot}25} = \underline{\cdot 6327 \quad \text{or } 63\tfrac{1}{4}\,\%}$$

$$\text{and efficiency of circuit only} = \frac{\text{H. P. delivered to motor}}{\text{H. P. generated}}$$

$$= \frac{17{\cdot}52}{20} = \cdot 876 \quad \text{or } 87{\cdot}6\,\%$$

there being 12·4 % of the generated H. P. lost in the wire.

Two actual cases may be quoted. (1) $4\frac{1}{2}$ H. P. was transmitted 8 miles through $\frac{5}{32}''$ telegraph wire, with a total efficiency of 30 %. (2) The dynamos having a resistance of 470 ohms, and the circuit 950 ohms, the line being 34 miles long, a total efficiency of 32 % was obtained by decreasing revolutions from 2100 at generator to 1400 at motor, the potentials dropping 2400 to 1600 volts, a method of working first advised by Siemens.

(*See p.* 929.)

Storage cells are objectionable for tramcar and locomotive driving on account of their great weight, 2 tons of cells being about the weight required for a 1-ton car. The following results are from an actual experiment with Faure accumulators:

35 cells of 95 lbs. each ...	=	$1\frac{1}{2}$ tons.	(*See pp.* 958 *and* 1118.)
H. P. absorbed in charging ...	=	1·558.	
Time of charging	=	22 hrs. 45 mins.	
Lost work in charging ...	=	34 %	
Chemically stored energy ...	=	66 %	
Recovered electric energy ...	=	60 % of 66 % = <u>39·6 %</u>.	

Figs. 367 and [illegible] are examples [illegible] the former [illegible] gear, the generator [illegible] to a line shaft [illegible] driven by [illegible]

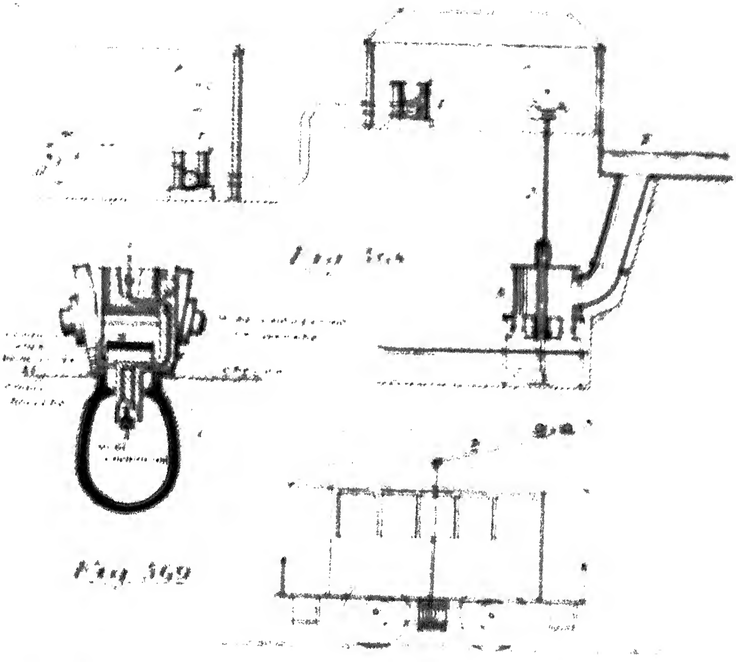


Fig. 368

Fig. 369

Examples in Electrical Transmission

line wire by a trolley a, through the 'fishing rod' c, and the return taking place by earth. [illegible] is the conduit system adopted by Mr [illegible], where the wire is underground, and the slot covered by a strong steel band. The latter is lifted by the little trolley, as it passes, to allow of the connection between circuit and motor. Electric travelling cranes are a recent development, the motors being placed on the travelling girders. (See pp. 916 sq. *Electr. Units and Measurement*; and p. [illegible], *Electr. Traction*.) [illegible]

Laws of Friction.—'Solid' friction (or the friction between solid surfaces) is here meant, in contradistinction to fluid friction. There are three laws, as follows:

The tractive force required to overcome friction :—

(1) Depends directly on the pressure between the surfaces in contact.
(2) Is independent of the extent of the pair of surfaces in contact, but (2*a*) increases in proportion to the number of pairs of surfaces.
(3) Is independent (at low velocities) of the relative velocity of the surfaces.

Further, the force depends on the co-efficient of friction (μ) for the particular materials, thus,

Tractive force $F_n = \mu P$ where P = total pressure.

COEFFICIENTS (μ) OF FRICTION AT VERY LOW SPEEDS. (MORIN.)

MATERIALS.	Method of Lubrication.						
	Dry.	Water.	Olive oil.	Lard.	Tallow	Dry soap.	Polished and greasy.
Wood on Wood...	·5	·68	...	·21	·19	·36	·35
Metal on Metal...	·18	...	·12	·1	·11	...	·15
Wood on Metal...	·6	·65	·1	·12	·12	...	·1
Hemp on Wood...	·63	·87	...	...	...	...	...
Leather on Iron...	·54	...	...	...	...	...	..
Leather on Wood	·47	...	...	...	...	...	·28
Stone on Stone...	·71	...	...	...	...	...	...
Stone on W.I. ...	·45	...	..	...	...	...	...
Wood on Stone...	·6	...	...	...	...	...	...

As μ is the trigonometrical tangent of the friction angle ϕ (*see p.* 560), the latter may be found as in Fig. 570, by dividing a base-line into tenths and setting up μ on a perpendicular from the mark 1, to the same scale. Thus, for dry metal, $\phi = 10°$, when $\mu = \cdot 18$. (*See Appendix II., p.* 868.)

Morin's experiments are not true for very heavy loads or at high velocities with much abrasion. For the first, Ball gives

$$F_n = \cdot 9 + \cdot 266\ P$$

for wood on wood, and the relation is set out in Fig. 571, the dotted line shewing the result of the ordinary formula with $\mu = \cdot 336$.

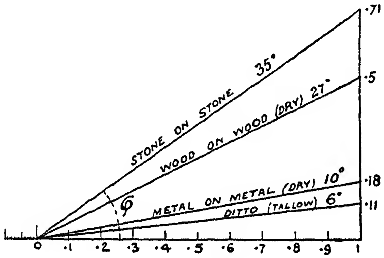


Fig. 570

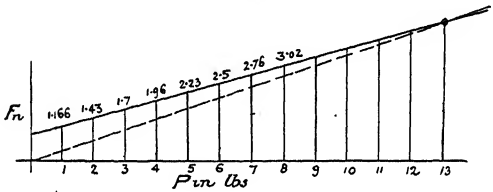


Fig. 571

As regards velocity, at the Brighton brake trials, 1878, the following results were obtained when the static coefficient was ·242.

Vel. ft. per sec.	μ between brake and wheel.	μ between wheel and rail.
80	·106	...
50	·153	·065
40	·171	·07
20	·213	·072
10	·242	·088
near rest	·242	·242

As there was probably considerable abrasion in these results, it is doubtful whether they should be accepted, further than generally, for pure friction. The second column shews that the wheels should not be allowed to skid when stopping the train.

If surfaces are thoroughly lubricated the frictional resistance is of a 'mixed' kind, being neither solid nor fluid. The following comparison is useful:

COMPARISON OF THE LAWS OF SOLID AND FLUID FRICTION.

Solid friction is :—	Fluid friction (gas or liquid) is :—
1. Directly as pressure.	1. Independent of pressure.
2. Independent of surface.	2. Directly as wetted surface.
3. Independent of velocity (at low velocities).	3. Directly as v at creeping velocities. as v^2 at moderate velocities. as v^3 at high velocities.

The **Friction of a Journal Bearing** was investigated by Beauchamp Tower for the Institute of Mechanical Engineers. The load was carried on one brass only, a top one, and the journal ran in an oil bath. The coefficient varied with the lubricant. With oil-bath lubrication F_n was independent of pressure, and $\mu \propto \frac{1}{P}$. In terms of velocity,

$$\mu = c\frac{\sqrt{v}}{p}$$

where c varies with the lubricant. Thus, when $v = 4$ and $p = 300$.

Lubricant.	c	μ	Lubricant.	c	μ
Olive oil	·289	·00192	Sperm oil .	·194	·00129
Lard oil	·281	·00187	Rape oil...	·212	·00141
Mineral grease	·431	·00287	Mineral oil	·276	·00184

With syphon lubrication $\mu = \frac{c_1}{p}$ where $c_1 = 2{\cdot}02$ for rape oil, and with pad lubrication $\mu = {\cdot}01$ for rape oil. The bearing seized when p rose above 600 lbs. (*See Appendix II., p.* 870.)

Friction of a Collar Bearing.—This was examined under the same auspices. Here the friction was nearer the 'solid' condition, the lubrication being less perfect. The pressure p varied from 15 to 90 lbs., and v from 5 to 15 ft. per sec. The coefficient was ·036 for ordinary loads, the usual formula being applicable. (*See p.* 871.)

Work Lost in Journal or Collar Friction.—R being outside or mean radius respectively,

$$\text{Work lost in foot pounds per m.} = F_n \times V = \mu P \times 2\pi R N$$

$$\text{and H. P. lost} = \frac{\mu P \times 2\pi R N}{33000}$$

Work Lost in Pivot Friction.—Following the method

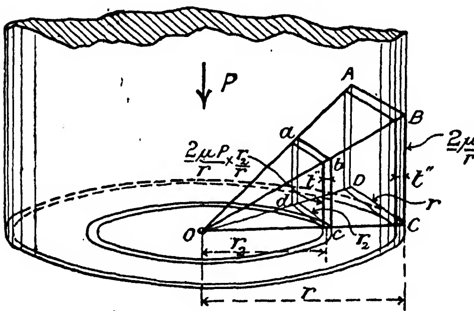


Fig. 572.

Work lost in Pivot friction.

of Fig. 371, let r be the pivot radius in Fig. 572. Assuming the pressure to be equally distributed,

$$\frac{P}{\pi r^2} = \text{pressure per sq. in., and } \frac{\mu P}{\pi r^2} = \text{force of friction per sq. in.}$$

$$\text{Total friction on any ring} = \text{unit friction} \times \text{area of ring}$$

$$\therefore \text{ Total friction on outer ring} = \frac{\mu P}{\pi r^2} \times 2\pi r \times t'' = \frac{2\mu P}{r} \times t''$$

$$\text{and} \quad \text{Total friction on ring } r_2 = \frac{r_2}{r} \times \frac{2\mu P}{r} \times t''$$

the resistance increasing gradually from O to B C. But the force must be multiplied by the arm to give the moment. The lamina A B C D represents the moment for the outer ring, being

$$\text{force}\left(\frac{2\mu P}{r}\right)\times \text{arm}\ (r)$$

Similarly $a\,b\,c\,d$ is the moment at ring r_2, and the pyramid volume will give the total moment, thus:

$$\underline{\text{Moment of friction}} = \frac{2\mu P}{r}\times r\times\frac{r}{3} = \frac{2}{3}\mu P r$$

If P and r are in lbs. and inches, the moment is in pound inches, and the distribution of pressure may be such as to reduce it to $\frac{1}{2}\mu P r$. Concentrating the total force at the outer ring, it will be $\frac{\frac{2}{3}\mu P r,}{r}$ and

$$\text{Work lost per m.} = \tfrac{2}{3}\mu P \times 2\pi R N$$
$$\text{which may decrease to } \tfrac{1}{2}\mu P \times 2\pi R N$$

Example 55.—Find H.P. lost in a footstep, whose dia. is 4″, total load 3000 lbs., revs. 100 per m., when $\mu = \cdot 06$. (Hons. Mach. Constr. Ex., 1887.)

$$\text{H. P. lost} = \frac{2\times\cdot 06\times 3000\times 2\times 22\times 2\times 100}{3\times 33000\times 7\times 12} = \underline{\cdot 38.}$$

Example 56.—Mean dia. of thrust bearing = 14″, screw thrust 40,000 lbs., and pitch 15 ft. $\mu = \cdot 003$, and 1000 miles are travelled in $3\frac{1}{2}$ days. Find H. P. lost in friction. (Eng. Ex., 1888.)

$$\text{Speed of vessel} = \frac{1000\times 5280}{3\cdot 5\times 24\times 60}\ \text{ft. per m.}$$

and as vessel travels 15 ft. per rev.

$$\text{Revs. per m.} = \frac{1000\times 5280}{3\cdot 5\times 24\times 60\times 15}$$

$$\therefore \underline{\text{H.P. lost}} = \frac{\mu P\, 2\pi R N}{33000} = \frac{\cdot 003\times 40000\times 2\times 22\times 7\times 1000\times 5280}{33000\times 7\times 12\times 3\cdot 5\times 24\times 60\times 15} = \underline{\cdot 93.}$$

The form of pivot surface may be flat, conical, spherical, or specially formed. If conical, $\frac{P}{\sin\alpha}$ must be substituted for P, where α = angle at cone apex. (*See p.* 507.)

Schiele's Pivot, Fig. 573, is generated by a *tractrix* revolving on its own axis. The curve is drawn as follows: Step off equal

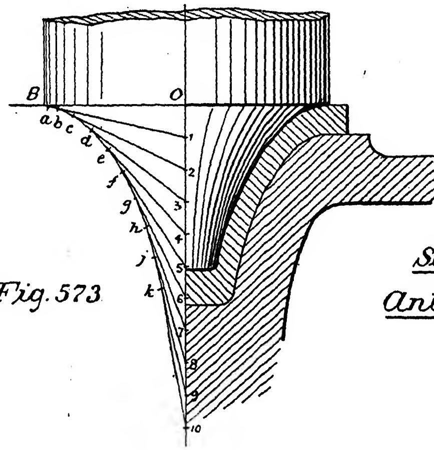


Fig. 573

Schiele's Anti-friction Pivot.

divisions 1 to 10; with radius OB and centre 1 set off 1*a* and join: with same radius and centre 2 set off 1*b* on 1*a* and join: similarly 3*c*, 4*d*, &c.; and then sketch the curve from B to K. This pivot wears equally on all rings, but wastes more energy in friction:

$$\text{Moment of friction} = \mu P r$$

or 50% in excess of a flat pivot.

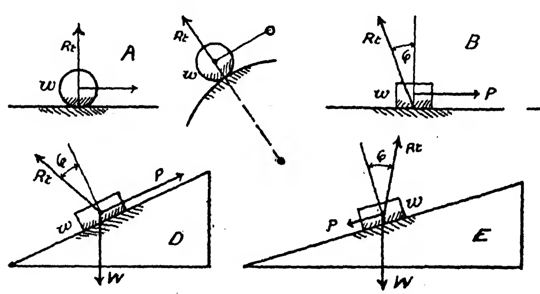


Fig. 574.

The **Limiting Angle of Resistance.**—If a weight rest on a perfectly smooth surface as at A, Fig. 574, the reaction is normal

to the surface, but if the surface be rough, the reaction is inclined to the normal by the friction angle, *in a direction away from the pull* P, and the latter must now be increased by F_n in order to move the body. If not on the point of sliding, the obliquity of R may be anything less, down to zero. Two cases are shewn for the inclined plane, P being directed up or down the plane, but its value may always be found by force diagram. In moving up the plane, total pull must balance gravity + F_n, but in moving down the plane must balance F_n − gravity.

Example 57.—A road engine weighs 12 tons. Find (1) tractive force of engine to pull 48 tons behind it on a level road, and (2) the load drawn up a 1 in 10 incline. Coefficient of traction = 150 lbs. per ton.

(1) $\underline{\text{Tractive force}} = (12+48)\,150 = \underline{9000 \text{ lbs.}}$

(2) $P \times \text{length} = W \times \text{height} \quad \text{and } P = W \times \frac{1}{10}$
Also R will be found to $= \cdot995\ W$

$$\text{Tractive force to balance gravity} = \frac{2240}{10} = 224 \text{ lbs. per ton.}$$

$$\text{Tractive force to overcome friction} = 150 \times \cdot995 = \underline{149\cdot25} \text{ ,, ,,}$$

$$\text{Total tractive force} = 373\cdot25 \text{ ,, ,,}$$

But the engine only exerts 9000 lbs.

$$\therefore \text{Total load on incline including engine} = \frac{9000}{373\cdot25} = 24\cdot11 \text{ tons.}$$

$$\underline{\text{and Load drawn exclusive of engine} = 24\cdot11 - 12 = 12\cdot11 \text{ tons.}}$$

Diminishing Friction by Lubrication.—Spongy metals like cast iron, brass, and white metal decrease frictional resistance considerably, but the best results are only obtained by the application of unguents.

Lubricants may be solid, as blacklead; semi-solid, as greases and fats; and liquid, as oils. 'Body' for support, and fluidity to avoid resistance, are both essential requisites, and a careful choice must be made between extremes. The following are the unguents used for various purposes:—

1. At low temperatures	Light petroleums.
2. For intense pressures	Graphite or soapstone.
3. Heavy pressures, slow speeds	Tallows.
4. Heavy pressures, high speeds	Sperm oil, heavy petroleums.
5. Light pressures, high speeds	Sperm oil, refined petroleums.
6. Ordinary machinery	Lard oil or tallow.
7. Steam cylinders	Tallow and heavy petroleums.
8. Metal on wood bearings	Water.

*Gumming or quick oxidation is to be avoided.

Lubricants are tested in about six ways:—(1) By chemical analysis; (2) for specific gravity; (3) for relative viscosity when new; (4) for gumming action; (5) for flashing and burning points; (6) generally, by testing machine.

Beaumé's hydrometer is shown at Fig. 535, being a glass tube a weighted by mercury at c, which floats in the liquid b to be tested, and the depth to which the stem sinks will show the relative density of b. Thus for

	Spec. Grav.		Spec. Grav.
Sperm oil	·881	Castor oil	·966
Olive oil	·916	Petroleum oil	·866
Lard oil	·915		

Viscosity may be observed by dropping the oil from a fine tube, and gumming by the apparatus in Fig. 536. The various oils are dropped through the holes a a, from the tube b, and as they travel slowly down a glass plate their positions on the scale are daily noted. Fig. 537 is a graphic rendering of some results, the slowing points being shown by circles and the stopping points by crosses. Flashing points are observed by heating the oil in a closed vessel, then lighting the gas when collected; burning points are where the whole oil takes fire. A low flashing point shows a dangerous oil, 73° F. being the minimum as fixed by law for commercial oils.

	Flashes.	Fires.	Burns.
Sperm oil	400°	485°	500°
Lard oil	475°	525°	585°

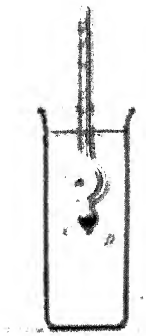


Fig 575

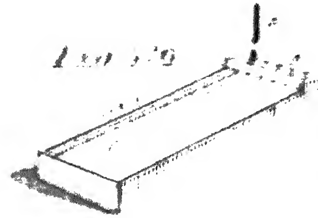


Fig 576

Fig 577

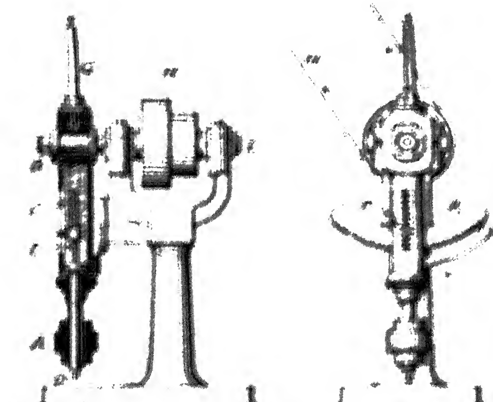


Fig 578

Prof Thurston's Universal testing Machine

The most effective test is obtained by machine, of which Professor Thurston's (Fig. 578) is probably the best. A is a pendulum hanging on the test journal B, whose brasses can be adjusted for any pressure by turning the screw D E against the spring C, while P shews the value, both totally and per square inch. The thermometer G indicates the temperature. The journal being rotated towards the right, the pendulum moves to the left, together with pointer F, and the scale K at once indicates the friction per sq. in. of journal, so that

$$\mu = \frac{\text{F's graduation.}}{\text{P's graduation.}}$$

Every five minutes during a test the revolutions, temperature, and graduations are noted, values of μ afterwards found, and the results plotted as curves wherever possible. In his 'railroad' machine, Prof. Thurston used a full-sized locomotive journal.

Lubrication.—The oil-bath gives the best result, but is rarely found in practice. The self-lubricating bearing, Fig. 579, is perhaps the next best, where the oil is lifted by the shaft collar and distributed by a wiper. The next in order is the oil pad, as contained in the locomotive axle-box, Fig. 580, the bush merely embracing the top half of the journal. Usually lubricators have to be fitted, and are then designed for the conditions. B, Fig. 270, p. 266, is a common syphon lubricator. The oil level being below the syphon-pipe, a piece of cotton wick is placed in the latter and hangs over in the oil. The fluid then rises by capillarity; and the wick is to be removed when the machinery is stopped, otherwise there is unnecessary loss of oil. Leuvain's needle lubricator, A, Fig. 581, is a glass vessel, filled with oil, closed by a wooden plug and inverted. Within the stopper a 'needle' fits freely, and the oil trickles down the latter only when vibrated by the shaft. If the dropping of the oil is to be observed and its regulation obtained, such a lubricator as the Crosby sight-feed at B, Fig. 581, may be adopted. When handle *a* is vertical, the valve *b* is raised, and adjustment given by the nut *d*; but when *a* is horizontal, *b* is closed, and the supply stopped.

A loose pulley may be fed with tallow by means of Stauffer's screw-down lubricator C. Oil would only fly away by centrifugal

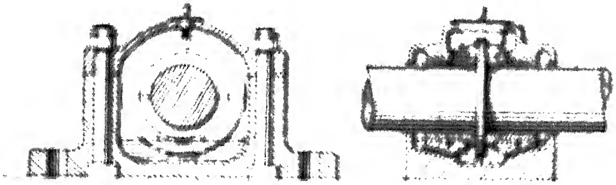


Fig. [illegible]

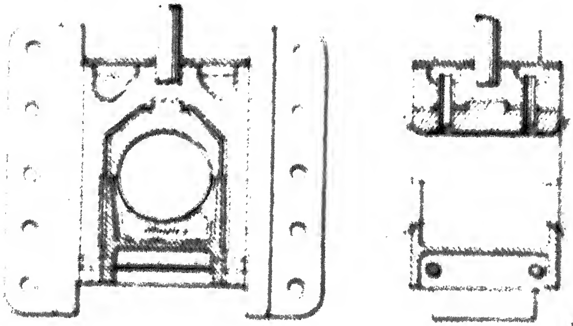


Fig. [illegible]

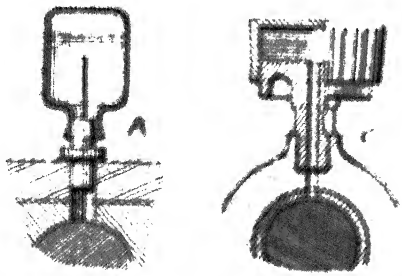


Fig. 381

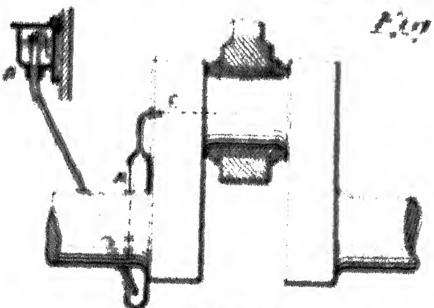
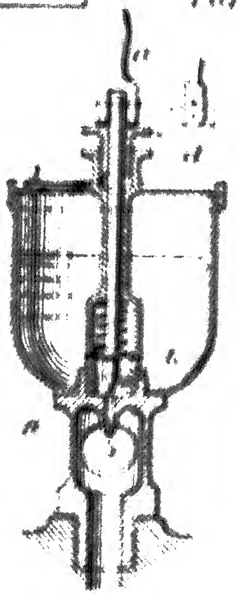


Fig. 382



Lubrication

action, and is therefore inadmissible. A [illegible] lubricator [illegible] used for steam cylinders, [illegible] described at p. 264 and Fig. 271. Fig. [illegible] shows a method of oiling an engine crank pin by centrifugal action. The oil being fed from a down the inclined tube, is caught by the [illegible] and whirled round, when it passes through the pipe [illegible] the crank pin.

Nothing but [illegible] [illegible] [illegible] [illegible] Grooves must be cut in the brasses from the lubricator pipe to the furthest ends of the brass, and more than one lubricator used for long bearings. In some cases small oil pumps have been adopted, but the oil tends to gum by exposure. (See *Appendix II*, p. [illegible].)

Contrivances for Diminishing Friction. The cart wheel A, Fig. 583, is the simplest example. Comparing with a sledge, the friction is reduced in proportion to the distance travelled by the sliding surfaces in each case, or as [illegible]. In small physical apparatus the anti-friction discs at B may be employed, the journal *a* resting on the wheel circumferences, and the sliding at *d d* being thus still further reduced. C is a coned bearing much used in clocks and watches, the work lost being here decreased by the adoption of a small diameter of rubbing surface; lathe centres form another example. A great reduction in friction is obtained when *rolling* is substituted for sliding contact. Examples are given on Figs. 584-86. Fig. 584 shows the 'live' rollers used to support the turret of an ironclad. They are tapered towards the centre A, being really bevel cones, and two light rings prevent them changing their proper relative position regarding their fellow rollers. Referring to Fig. 585, the weight Q compresses the rollers, in the manner shown exaggeratedly at the dotted arcs. The rollers will tend to turn round the fulcra marked, and the equation of moments will be

$$N(b + b_1) + W b_1 = T \times 2r$$

But $P = 2T$ and $Q = 2N$ $\therefore$ $P = \dfrac{Q(b + b_1) + 2Wb}{2r}$

If more rollers are used, let n = number of rollers: then

$$P = nT \text{ and } T = \frac{P}{n} \qquad Q = nN \text{ and } N = \frac{Q}{n}$$

The equation becomes:

$$\frac{Q}{n}(\delta + \delta_1) + W\,\delta_1 = \frac{P}{n} \times 2\,r$$

$$\therefore\ P = \frac{Q\,(\delta + \delta_1) + n\,W\,\delta_1}{2\,r}$$

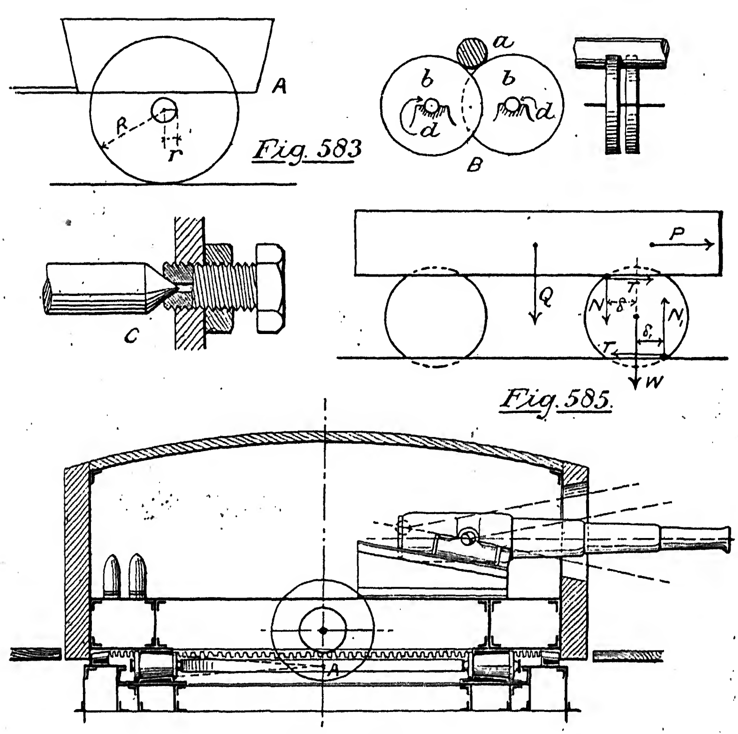


Fig. 583

Fig. 585.

Fig. 584.

δ is found by experiment, and

δ = ·036″ for rollers of wood 3′ to 4′ long.
δ = ·072″ for rollers of wood 1′ long.
δ = ·016″ to ·018″ for rollers of iron 5″ or 6″ long.

Prof. Osborne Reynolds shows that the act[illegible] as we have supposed, and Prof. [illegible] gives [illegible]

$$F = \frac{[illegible]}{r}$$

where k = [illegible] for hard [illegible]
k = [illegible] for softer materials
k = [illegible] for [illegible]

which is the same as formula [illegible] weight of roller. Rolling friction is not [illegible] by lubrication. (*See Appendices II and III, pp.* [illegible] *and* [illegible].)

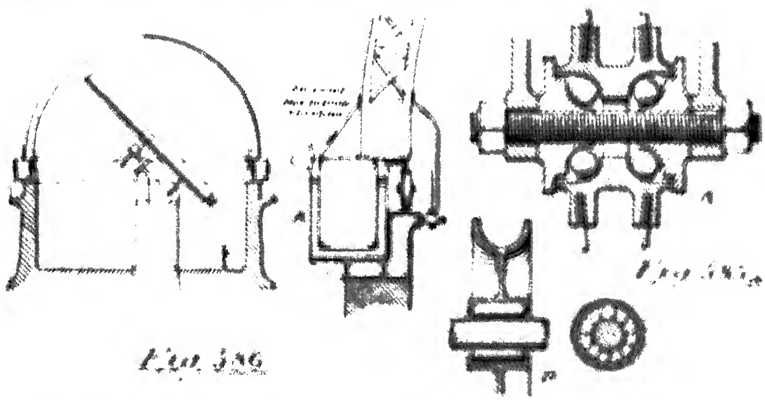


Fig. 586.

Ball and roller bearings are shown at A and B respectively, Fig. 586. The former is an excellent arrangement, but the latter cannot be adjusted after wear. Knife edges, Fig. 551, form an example of statical contact. Fig. 586 shows also a case where fluid friction has been substituted by Eiffel with advantage. The observatory dome at Nice was floated in an annular tank, whose section is given, the liquid having a specific gravity of 1½. The moving load was 95 tons, but could be turned by one man in four minutes. The live rollers were not a support but only a steadiment against wind. (*See Appendix II., p.* 918.)

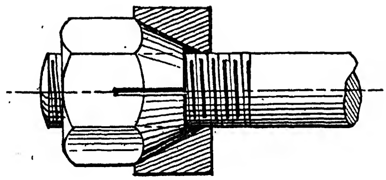
Uses of Friction. Very often friction is a positive advantage; such cases we will now discuss. Fig. 587 is one of many *lock nuts*, the grip being obtained by the compression of the split nut. *Friction clutches* provide disengagement for shafts or pulleys

while the machinery is in motion. There are many examples in the market, all possessing some advantage or another. That in Fig. 688 combines [illegible] principle with Hall's and Stuart's adjustment. a is a loose pulley on shaft b, and the boss c is keyed to b, is keyed to the shaft, and carries the right and left handed screws e e. Levers f f, [illegible] upon d d, are connected to links m m to the sliding [illegible] n, worked by lever, so it follows that when n is moved to the right and the screws rotated, the gripping shoes are pressed against the boss, thus connecting the pulley with the shaft. s s are small worm spindles for adjusting the levers to suit wear of shoes. A clutch like the above should be symmetrical in design, so as to be in perfect balance; the shoes should not rest upon the boss when out of gear, and there should be good adjustment. [illegible]

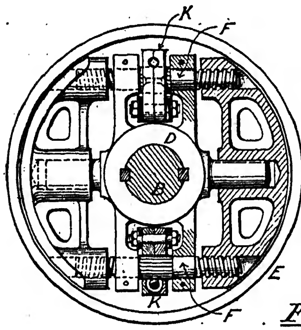
Weston's clutch, Fig. 689, is designed on the principle of multiple gripping surfaces (see Ans. to Q. [illegible]). It is now adopted mostly as a safety appliance, allowing wheels to slip when a shock comes upon them, and thus avoiding breakage. The example shown is the elevating gear for a large gun, the pinion on the right gearing with a rack on the gun, and the connection from a to b being made by means of the clutch. Steel discs d d fit on the hexagon c, but are free regarding the worm wheel, and gun metal discs f f are keyed to the wheel but free on the shaft. When nut c is tightened, the discs are gripped, and a is connected to b, but when c is released, the wheel is free. m are spring washers.

Brake straps are a means of absorbing power by friction, dissipating it as heat. Fig. 690 represents a traction engine having a brake drum a securely keyed to its hind axle c. The iron strap b, sometimes lined with wood or leather, encircles the drum, and is tightened whenever the lever d is raised by the screw. Brake blocks, with toggle gear, are shown at Fig. 465.

Friction plays an important part in causing a grip between a locomotive driving wheel and the rails, and the weight of the engine should be sufficient to prevent slipping when starting the load. The resistance of a train to direct pull is from [illegible] to [illegible] and even 30 lbs. per ton, which gives the tractive force required. The resistance to slipping is found from the following table:—



The Gripper Lock-nut
Fig. 587.



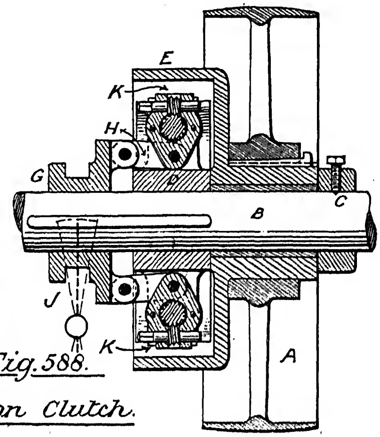


Fig. 588.

Disengaging Friction Clutch.

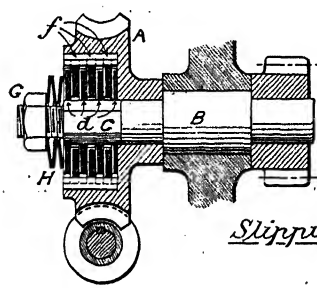
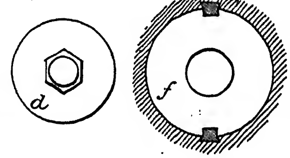


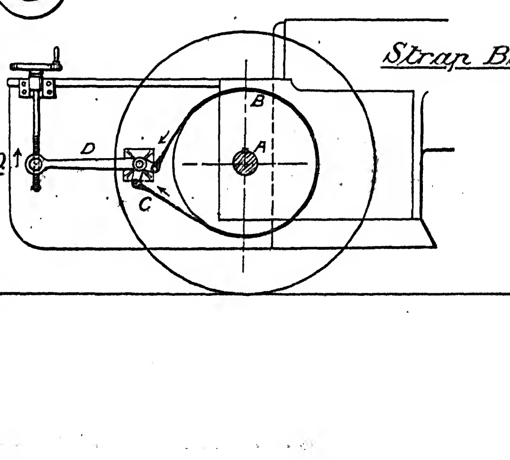
Fig. 589



Slipping Friction Clutch.

Strap Brake.

Fig. 590



ADHESION OF LOCOMOTIVES

(PER TON OF LOAD ON DRIVING WHEELS.*)

Rails very dry	600 lbs. per ton
Rails very wet	550 ,, ,,
Average weather	450 ,, ,,
Greasy rails	300 ,, ,,
Frosty weather	200 ,, ,,

110.) **Friction Gearing** transmits power without jar, and will slip under shock. The forces transmitted are, however, limited. Referring to Fig. 591,

$$F_t = \mu P \text{ and } P = \frac{F_t}{\mu}$$

Taking $\mu = .25$ for leather on iron, pressure on bearings $= \frac{F}{.25} = 4$ times the force transmitted (see A) and in right angled bevel gear $= \frac{4}{\sqrt{2}}$ times the force transmitted (see B).

To avoid bearing pressure, Prof. Jenkin invented his 'nest gearing,' which is shown in Fig. 592, transmitting power between engine shaft A and dynamo B. To obtain adjustment for the intermediate wheels r_2 r_3 r_4, the shafts A B are out of line, and the intermediate studs fixed to a plate with curved slots. The disadvantage of the gear lies in its having six compressed surfaces instead of two.

In Fig. 593, C shows examples of Robertson's wedge gearing, and D, a more recent design of friction gearing, has plates of leather on edge forming the driving surface, the followers being smooth cast iron. In both cases grip must be obtained by causing pressure on the bearings, either by spring as at Fig. 592, or by weight or changing eccentric bush as at C, Fig. 593.

Efficiencies of Machines. The frictional loss in a machine could be investigated for every kinematic pair, but in a large machine this would be cumbersome, and considering the variations in the value μ, very probably inaccurate. The engineer prefers then to make experiments upon existing machines and to

* Every coupled wheel is a driving wheel.

keep a list of results ... [illegible]
... of P to W, we ... the ... [illegible]
... which is the efficiency of the ... [illegible]
could be taken as a ... [illegible]

Fig. 591.

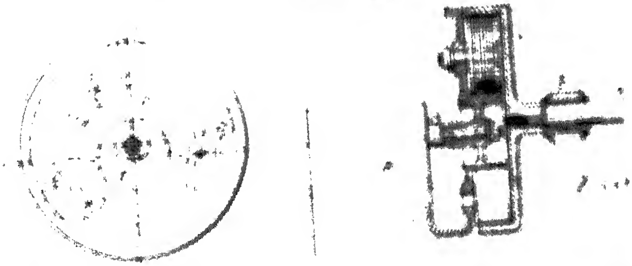


Fig. 592.

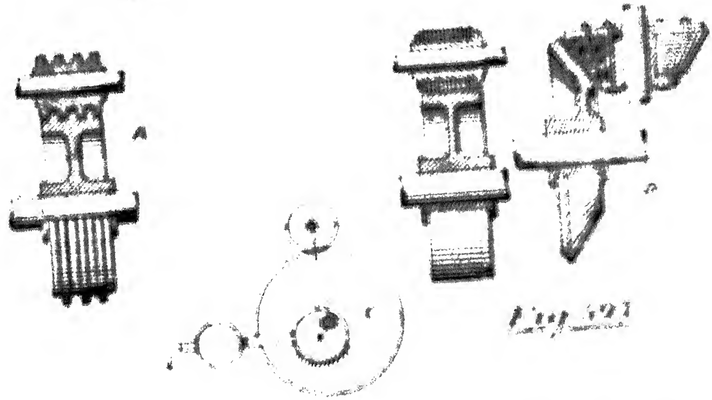
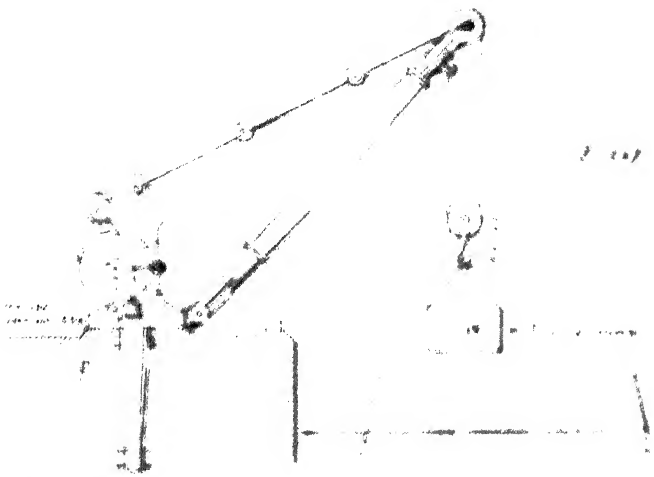


Fig. 593.

... meter to be applied, we will, however, consider the ... in Fig. 594.

Commencing by measuring the motions of W and P, we find that while W moves one inch P moves 224 ins., so the velocity ratio of P and W is 224 : 1. Calculate then the theoretical values

[illegible]



that will just cause a slow but uniform [illegible] after starting by hand; note the results and tabulate as follows:

[illegible]

Load W in tons	Weight [illegible] to just overcome load	Efficiency
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
unloaded	[illegible]	[illegible]

Next plot these figures as in Fig. 595, the horizontal
shewing W, and the vertical ordinates the corresponding valu
P ; O D being the theoretical, and A B the practical profiles, w

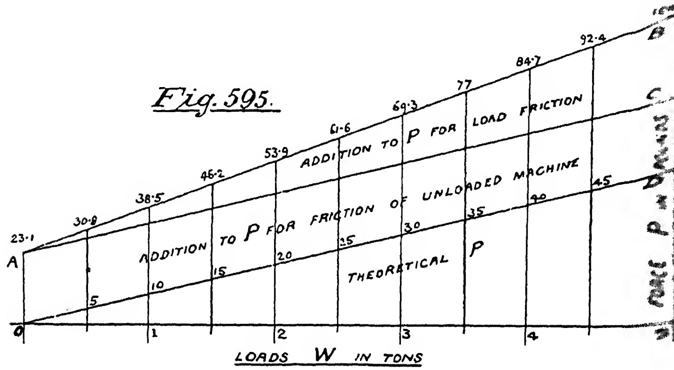


Fig. 595.

are both straight lines. Draw A C || to O D. Then at any ordi
except O A the total P consists of :

1. Force to overcome load, neglecting friction.
2. Force to overcome friction of unloaded machine.
3. Force to overcome friction due to load.

(2) being a constant quantity as shewn between lines A C, O D

Then, if w be the equivalent weight of the unloaded mac
causing friction,

$$\begin{array}{ccccc} & (1) & & (2) & (3) \\ P = & \frac{1}{224}W & + & \mu w & + \mu W \end{array}$$

$$\text{From (3) we find } \mu = \frac{F_n}{W} = \frac{BC}{5 \times 2240} = \cdot 00687.$$

Supposing μ a constant throughout the machine,

$$P = \frac{W}{224} + \cdot 00687\, w + \cdot 00687\, W$$

and as $CA = 23 \cdot 1 = 3362 \times \cdot 00687 \quad \therefore \quad w = 3362$ lbs.

$$\text{Efficiency per cent} = \frac{\text{power at [illegible]}}{\text{power put in}} = \text{[illegible]}$$

When W = [illegible] tons, efficiency = [illegible] = [illegible] nearly;

and when W = [illegible] tons, efficiency = [illegible] = [illegible]% nearly;

which shows the advantage of working a machine near its full load. (See *Appendix II., p.* [illegible], *and Appendix II., p.* [illegible].)

Dynamometers are best used when the power given to or by a continuously running machine is to be ascertained. The method just described might well be employed to measure *starting* loads, but *working* loads are often much less, and the above treatment is not then admissible. Dynamometers shew the load supported with any speed of revolution, and the latter is measured by a counter. Absorption dynamometers abstract the work while measuring it and dissipate it as heat, while transmission dynamometers pass it on unimpaired, absorbing only an inappreciable amount in frictional loss.

An *Absorption Dynamometer*, or Brake, is shewn in Fig. 596. The engine whose power is to be measured has its crank coupled to *A*, and revolves in the direction of the arrow. Belt driving should be avoided on account of the loss in slip. The drum *B* carries a brake strap lined with wood, which is tightened at *D*. Sufficient standing weight is hung at *W*, whose rise is prevented by the stop *P*, after *D* has been screwed up to just support the average load, the self-acting lever preventing any important rise or fall of the points *E*, *D*, which must be level with drum centre. As *E* would travel further than *F*, a right-handed rise of the strap will slacken the latter, and a left-handed dip tighten it, thus preserving the original position. The spring balance *S* conveniently measures small deviations of load, and the weight supported will be $(W - S)$, the radii R, R, being equal. Then work absorbed per rev. = weight × distance travelled = $(W - S)$ lbs. × $2\pi R$ feet (if the pressure on stop *P* be inappreciable); and

$$\text{H.P.} = \frac{(W - S) \times 2\pi R N}{33000}$$ (See *Appendix II., p.* 833.)

Friction is only the *medium* for absorption, and does not enter into the calculation. The arrangement at K permits adjustment for various motors.

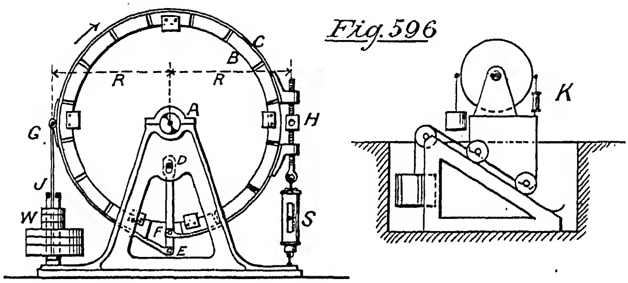


Fig. 596

Absorption Dynamometer.

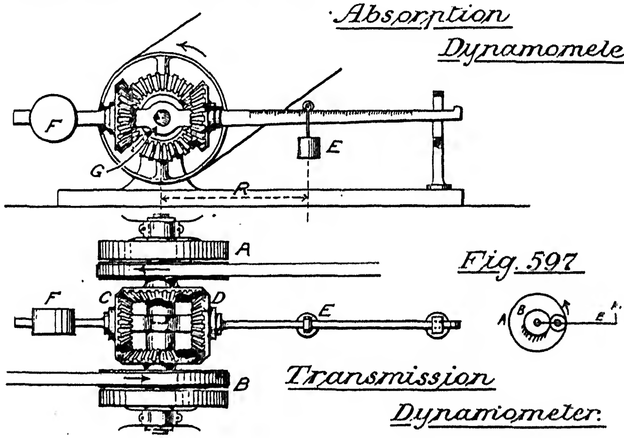


Fig. 597

Transmission Dynamometer.

White's Transmission Dynamometer is represented in Fig. 597. A is the motor shaft, and B that of the driven machine. As A turns left-handed, the arm F E is held back by the weight E, and thus B is turned to the right. Supposing the arm *were* carried round, *no work* would be given to B, which would be stationary, but E's rotations would be half those of A (*see Fig.* 526). The load supported on A would therefore be half that on E (at equal radii). But although the power be taken off at B, A and E have yet the same relation, so that

work transmitted = load on A × distance travelled on A
and load on A = half that on E,

$\therefore$ work per min. $= \frac{E}{2} \times 2\pi R N$ and H. P. $= \frac{E \pi R N}{33000}$

F counterbalances the lever weight.

List of Efficiencies.—Efficiencies in various cases, as found by the methods previously described, are as follows:

Cranes worked by spur gearing...	30% to 60%
Worm gearing (indifferently constructed)	30%
Worm gearing (very carefully constructed)	90%
Weston pulley block, well greased	30% to 40%
Screw jack	15% to 35%
Cornish engine (Brake H. P. ÷ Indic. H. P.)...	35% to 60%
Other engines (Marine, Loco, Gas, Oil, &c.)...	75% to 85%
Undershot water wheels...	25% to 30%
Overshot water wheels ...	70% to 75%
Breast wheels (Poncelet floats)	60% to 65%
Pelton water wheel	80% to 90%
Turbines (full sluice)	60% to 80%
Hydraulic press (neglecting pump)	98% to 99%
Hydraulic jack with pump	77%
Pumps (piston)	78%
Hydraulic accumulator ...	91%
Hydraulic lift, working rapidly	50%
Hydraulic cranes, all losses taken	55%

Mechanical efficiency of engines, not varying appreciably with load, is often found by comparing an indicator diagram taken 'running light,' with that under working conditions. (*See Appendix IV., p.* 962.) (*See pp.* 874 *and* 1125.)

We will close this chapter with a few comparisons of power transmitters:

Comparisons of the Advantages and Disadvantages of Transmitting Power by Various Methods.

Advantages.	*Disadvantages.*
1. Linkwork.	
Useful in modifying power and obtaining special motions, as with valve gear, parallel motions, &c. Coupling rods a case of pure transmission.	Dead points often occur, to be overcome by force or chain closure. Will only transmit over very short spaces.
Frictional loss slight	... but depends on number of joints.

Advantages. *Disadvantages.*

6. [illegible]

Useful in connection with belt and spur gearing at a distance from central motor to machines.

Practically noiseless.

Will transmit power moderate distances only, unless used in conjunction with belt gearing and [illegible] shafts.

[illegible]

7. Spur Gearing.

For positive transmission.

A good modifier of power.

No pressure on bearings due to wedge action, if teeth are well formed. Only pressures are due to weight of wheels and unbalanced moment. The former often considerable, but the latter not present in all cases.

Frictional loss [illegible] per cent single gear [illegible]%, double gear [illegible]%, treble gear [illegible]%.

Not suitable for long distances except in conjunction with line shafting and bevel gearing.

Noisy, especially if teeth are rough.

Teeth break [illegible] [illegible] interposed. Bearings [illegible]

8. Bevel Gearing.

Assists shafting in turning corners, and modifies at the same time, if required.

Is therefore useful in connecting parallel shafts considerably separated. Should not be used if belting or spur gearing will serve (see Fig. 276).

Frictional loss quite as great as with spur gearing; the teeth are difficult to cut by machine, and are rarely well formed.

Noisy for similar reasons.

Oblique pressure on bearings.

9. Worm Gearing.

Gives a high velocity ratio with few parts.

Non-reversible if the ratio be greater than 8 : 1, and therefore serves as a safety gear in cranes and such appliances.

Practically noiseless.

Frictional loss 10% to 50%. The former result has only been reached with exceptionally well cut teeth, and the worm in an oil bath. 30%, is a good average.

Machine cut teeth (p. [illegible]) should be used wherever possible.

(See p. [illegible].)

10. Screw Gearing.

Comes under the last head.

6. Belt Gearing.

Advantages.	*Disadvantages.*
Useful in connection with shafting as a distributor and modifier with comparatively few parts.	Frictional loss principally in the line shafting: about 25% to 50% in a shop system.
Easily started and stopped.	Large belts with heavy pressures are expensive to maintain.
Practically noiseless.	
Very convenient for bridging reasonable distances.	
Large pull on bearings, but in well-lubricated bearings friction does not depend on pressure.	
Slip an advantage in case of shock.	Slip a disadvantage where exact velocity ratio is required.

7. Cotton-rope Gearing.

Advantages.	*Disadvantages.*
For fairly long-distance driving in mills, and for travelling cranes.	Frictional and other losses probably somewhat larger than with belt gearing, due to heavy pullies and fly-wheels.
Better grip than belts, due to wedge-groove pulleys.	Working speeds being high, rope tension is increased 50% by centrifugal force: but bearing pressures are not thereby affected.
Quite noiseless.	
Separate driving to the various floors of a mill occasions less loss of time in breakdowns.	
Small liability to break down also.	

8. Wire-rope Gearing.

Advantages.	*Disadvantages.*
Suitable for very long distances, say for several miles, when relays are adopted. Cases quoted in text.	Frictional and other losses 22% per mile, not including motor and machines: lesser and greater distances in proportion.
If moderate speeds be employed, little increased tension from centrifugal force.	

9. Pitch-chain Gearing.

Advantages.	*Disadvantages.*
As useful as belt driving in decreasing the number of parts while modifying the power: but gives at the same time positive transmission, and may be used with heavy loads.	Frictional loss depends very much on design and manufacture, and probably varies from 5% to 30% in a pair of wheels: there being two sets of friction surfaces, not including the journals.
Adapted for high as well as low speeds if well made, but the former should go with light pressures.	Increase of pitch after wear causes excessive friction and bad working.

10. Friction Gearing.

Advantages.	*Disadvantages.*
Almost noiseless and non-vibrating. Advantage of slip when shocks are received. Useful for high speeds.	Frictional loss about the same as for belt driving to shafting, but comparatively small with one pair of wheels. Unequal wear. Large pressure on bearings; decreased in nest gearing.

11. Compressed-air Transmission.

Advantages.	*Disadvantages.*
Of great value for long-distance transmission in close workings. Better than hydraulics when high speeds are required in piston motors.	Loss by cooling varies from $70^{\circ}/_{\circ}$ under bad conditions to $20^{\circ}/_{\circ}$ with re-heating and air injection. Loss per mile by friction about $5^{\circ}/_{\circ}$.

12. Hydraulic (Water Power) Transmission.

Advantages.	*Disadvantages.*
Suitable for long distances. More especially useful for intermittent demand in power distribution, and the concentration of immense power by aggregating storage. Leakage slight. Inertia an advantage sometimes, as in riveting machines. Losses slight if low velocities are taken, say $15^{\circ}/_{\circ}$ in usual machines; $5^{\circ}/_{\circ}$ per mile due to friction in pipe.	Unsuitable for continuous work. Uneconomical with high velocities and reversible motion, on account of shock due to inertia. (Damage obviated by relief valves.) Velocity should be kept down to 4 or 6 ft. per second usually, and slow moving rams adopted, necessitating multiplying gearing. Piston engines run at 60 or 80 ft. per m. but are usually wasteful.

13. Electrical Transmission.

Advantages.	*Disadvantages.*
Especially suitable for long distances. Wires may be conducted in any direction. No moving parts in line of transmission. Easy subdivision of power. May be stored by secondary cells.	Loss in line varies as the square of the current used (C^2R): hence high voltage is adopted for long lines, giving an economic loss of from $5^{\circ}/_{\circ}$ to $40^{\circ}/_{\circ}$ in the line. Loss in dynamos from $5^{\circ}/_{\circ}$ to $20^{\circ}/_{\circ}$ each, of the energy intrusted to them. Storage cells, being heavy, are not really suitable for transportation purposes. Loss in charging *and* discharging, say $50^{\circ}/_{\circ}$

(*See Appendix II., p.* 875, *and Appendix III., p.* 928.)

CHAPTER X

ON HEAT AND HEAT ENGINES

It is customary, in dealing with any branch of Natural Science, to collect known facts from time to time, in order to devise a general *Theory* of explanation, which may serve as a basis for further investigation. Two such theories we shall now consider.

The **Molecular Theory** supposes that matter is discontinuous, its limit of mechanical divisibility being the minute particle termed a *molecule*. Chemists show that some molecules are compounded, being further divisible by chemical means into *atoms*.

The **Dynamical Theory of Heat** teaches that heat is not a substance, but a *condition* of matter, being a "pendulum" motion of the molecules, never entirely absent, even during extreme cold, but increasing with the intensity of heat, the latter being, in fact, due to the motion. In solids, the molecules are very close together and their excursions small, being limited by mutual attraction or *cohesion*; in liquids, they glide about and change positions by but slight external force; while in gases, the heat energy overcomes the cohesive or molecular forces, and the particles fly out to any distance when allowed to do so.

Black taught the *caloric* or material theory of heat in [illegible], but Rumford and Davy, in [illegible], showed that heat could be produced mechanically by solid friction, and thus proved it identical with motion. A little thought will suggest many cases where work and heat are interchangeable.

Transfer of Heat. When a hot and cold body are placed in juxtaposition, heat passes from the former to the latter till

both have equal temperatures. Such transference may occur by radiation, conduction, or convection.

Radiation is the passage of heat between substances not in contact, *without at the same time raising the temperature of the intervening medium.* Thus a fire may heat surrounding solids, and the air receive its heat from the solids in turn. To explain radiant heat, a fluid of infinite tenuity is imagined, called *the Ether*, filling space and the interstices of matter, and transmitting radiant heat, by wave motion, without increasing molecular motion. If, however, the undulations be arrested, the energy is absorbed as molecular motion, and becomes apparent in the arresting body as heat. Radiation is an aid to heat dispersion, as in heating apparatus, but a disadvantage with boilers and steam cylinders, there causing loss. Good radiators must therefore be adopted in the former, and bad ones for the latter case. Good radiators are good absorbers, to an equal degree, and reflecting power is the exact inverse of radiating power.

Relative Value of Radiators.

Substance.	Relative Radiating Value.
Lampblack or soot	100
Cast iron, polished	26
Wrought iron, polished	23
Steel, polished	18
Brass, polished	7
Copper, polished...	5
Silver, polished	3

Conduction is the transfer of heat by contact, molecular motion being then directly caused. Heat is thus transmitted through the thickness of a furnace tube. There are good and bad conductors, the former being chosen for fireboxes, other properties being suitable. (*See Appendix II., p.* 876.)

RELATIVE VALUE OF GOOD CONDUCTORS.

Substance.	Relative Conducting Value.
Silver	100
Copper	73·6
Brass	23·1
Iron	11·9
Steel	11·6
Platinum	8·4
Bismuth	1·8
Water	·147

Bad conductors are of value for clothing boilers, steam cylinders and pipes, &c. (*See p.* 902.)

RELATIVE VALUE OF BAD CONDUCTORS (OBSTRUCTORS).

Substance.	Relative Obstructing Value.
Silicate cotton or slag wool ...	100
Hair felt	85·4
Cotton wool	82
Sheep's wool	73·5
Infusorial earth	73·5
Charcoal	71·4
Sawdust	61·3
Gasworks breeze	43·4
Wood, and air space	35·7

Convection is a means of transmitting heat to liquids and gases. A flask of water being placed over some heat source, the lower or heated portion of the water becomes lighter and rises to the surface, up the vertical centre-line, only to become cool again and flow down the sides to the bottom. Thus are continuous 'convection' currents formed, which soon distribute heat throughout the liquid. Similarly also is the air of a room

heated: the fire, near the floor, rarefies the immediately surrounding air, which rises to the ceiling and falls again when cooled against the walls. Water, being a bad conductor, cannot well be heated by any but the convection method, hence the adoption of a low position, in a boiler, for the fire-grate.

Expansion is the result of the application of heat to all bodies, whether solid, liquid, or gaseous; the first being least and the last most expansible. Many examples may be suggested of the application of this law, some useful and some detrimental. Shrinking of gun coils is of the former type, while the endlong clearance between rail lengths of the permanent way avoids the injurious effects of the summer heat. Fig. 327 shews how work might be done by the expansion of solids. Water, between 32° and 39·1° F., is an exception to the law of expansion; during that period it contracts as the temperature increases. Cast iron also expands when solidifying in the mould, and bismuth and antimony follow the same rule; gold, silver, and copper contract.

Measurement of Heat.—We proceed to measure *intensity* and *quantity* of heat, bearing in mind, however, that heat is not a substance but a form of energy.

Temperature is a measure of the *intensity* of heat, being registered on a thermometer or pyrometer. Thermometers are based on the expansion of liquids or gases in a glass bulb, which then rise in a capillary stem from which air has been exhausted. Mercury or alcohol are the usual liquids, the former for ordinary and comparatively high temperatures, and the latter for very low temperatures: the boiling point of mercury being very high, and the freezing point of alcohol very low. The freezing and boiling points of water, under atmospheric pressure, being unchangeable, are first marked on all thermometers, after which the graduations are spaced according to one of the following methods:

Thermometer.	Divisions between Freezing and Boiling.	Freezing Point.	Boiling Point.
Fahrenheit ...	180	32°	212°
Centigrade ...	100	0°	100°
Réaumur ...	80	0°	80°

Réaumur divisions are adopted in Russia; those of the Centigrade by scientists and the Continental public; while Fahrenheit divisions, being used by English engineers and the English speaking public generally, will therefore be adopted in this work, and the Fahrenheit degree be looked upon as the *unit of intensity*. Centigrade readings can be translated into Fahrenheit and *vice versâ*, by the following simple formulæ:

$$F^\circ = (C^\circ \times \tfrac{9}{5}) + 32 \quad \text{and} \quad C^\circ = (F^\circ - 32)\tfrac{5}{9}.$$

Pyrometers are required to measure excessive temperatures, such as those of furnaces; they are discussed on page 587.

Air thermometers are of advantage in experiments of great delicacy, because small increase of heat will cause large expansion of air. The instrument is usually laid horizontally, and has a small index of coloured sulphuric acid, as at C, Fig. 601, which is moved along the tube by the expanding air, the end B being open to the atmosphere. The reading is considerably affected by change of atmospheric pressure, so the barometer reading must always be taken, and a correction made to standard pressure. The expansion of gases is more perfect than that of liquids. (*See pp.* 587 *and* 1126.)

Quantity of Heat.—More or less heat motion may exist in a body, depending on mass, heat capacity, and temperature. *The British Thermal Unit* (B.T.U.) *is the amount of heat required to raise the temperature of a pound of water through one Fahrenheit degree*, the water being near its greatest density 39·1° F. This unit represents an amount of energy equal to about 772 foot pounds.

Specific Heat.—But some bodies have greater capacity for heat than others, that is, weight for weight, will absorb more heat for a definite rise of temperature. Taking capacity for water as 1, the relative capacity of another substance called its *Specific Heat*, is therefore *the amount of heat in thermal units required to raise the temperature of a pound of the substance through one degree F.* Bunsen's ice calorimeter has been used to determine various specific heats, but we shall describe the *method of mixture*, which is precisely the same in principle. The body, being regularly heated in a bath of steam, is removed, and put in a vessel containing a measured weight of water at a certain temperature.

When the body and the water are in thermal equilibrium, the final temperature T° of the mixture is taken. Then, the heat lost by the body divides itself between the water and the casing, so

$$\begin{array}{ccc} \text{Heat lost by body} & = & \text{Heat gained by water \& casing.} \\ \text{weight} \times \text{sp. ht.} \times \text{fall of temp.} & = & \text{weights} \times \text{sp. hts.} \times \text{rise of temp.} \\ ws\,(t^\circ - T^\circ) & = & (w_1 s_1 + w_2 s_2)\,(T - t_1), \end{array}$$

where w w_1 and w_2 are the weights in lbs., and s s_1 and s_2 the specific heats of the body, the water, and the casing respectively : s_1 being unity. The value $w_2 s_2$ is known as the 'water equivalent' of the vessel. The first temperature of the water is t_1° while t° is that of the body after steaming; and the changes are shewn graphically at A, Fig. 598. Inserting known values, that of

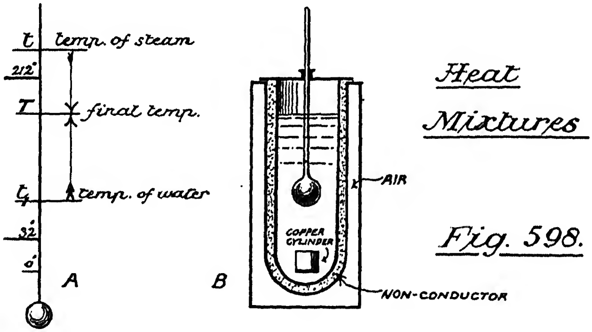


Fig. 598.

s may be found, the following table being obtained by this and other methods :—

Specific Heats of Various Substances.

Water at 39·1° ...	1·00	Wrought iron ...	·113
Water at 212° ...	1·013	Steel	·116
Ice at 32° ...	·504	Copper	·095
Steam at 212° ...	·48	Coal	·24
Mercury ...	·033	Air	·238
Cast iron ...	·13	Hydrogen	3·404

Example 58.—Find the specific heat of copper from the following data:—Half a pound of copper is heated to 212°, and being plunged into a pint (20 oz.) of water at 60° contained in a wrought iron vessel weighing 4 oz., raises the temperature of the latter to $65\frac{3}{8}$°.

$$s = \frac{(w_1 s_1 + w_1 s_2)\,(T^\circ - t_1^\circ)}{w\,(t^\circ - T^\circ)} = \frac{(1{\cdot}25 + {\cdot}25 \times {\cdot}113)\,(65\frac{3}{8} - 60)}{{\cdot}5\,(212 - 65\frac{3}{8})} = {\cdot}0938$$

Pyrometers, for measuring very high temperatures.—Wedgewood's and Daniell's, based on expansion of solids, are now obsolete. Siemens' electric pyrometer measures the resistance of a circuit, which varies directly as the temperature of the wire. Wilson's and Siemens' water pyrometers depend on the method of mixtures. A cylindrical vessel of sheet copper, clothed with felt to retain heat, is provided with a cover and thermometer (see B, Fig. 598). A small solid cylinder of copper, of known weight, being placed in the furnace whose temperature is required, is shortly removed, and plunged into the water of the pyrometer, when the latter is closed. The final temperature of the pyrometer water being observed, that of the furnace can be deduced. (*See Appendix II., p.* 876.)

Example 59. — Find a flue temperature by water pyrometer from the following data:—Quantity of water = 1 pint, its first temperature 65°; weight of copper cylinder = $4\frac{1}{4}$ oz.; final temperature of water = 72°: water equivalent of vessel = ·38 (lb. F°)

$$w\,s\,t^\circ - w\,s\,T^\circ = (w_1 + {\cdot}38)\,(T^\circ - t_1^\circ)$$

$$t^\circ = \frac{(w_1 + {\cdot}38)\,(T^\circ - t_1^\circ) + w\,s\,T^\circ}{w\,s} = \frac{(1{\cdot}25 + {\cdot}38)\,7 + ({\cdot}265 \times {\cdot}095 \times 72)}{{\cdot}265 \times {\cdot}095} = 528$$

Expansion of Gases.—Two laws govern the varying volume of a gas, according to whether temperature or pressure be kept constant.

The first law of gas expansion, discovered by Boyle in 1662, and verified by Marriotte in 1676, states that *the volume of a given portion of gas varies inversely as its pressure, if the temperature be constant.* Shewn by symbols:

$$V \propto \frac{1}{P} \quad \text{and} \quad PV = \text{a constant.}$$

The relation of P and V is given by diagram in Fig. 599, the ordinates PP_1 of the curve representing pressures, and the

abscissæ V V_1 corresponding volumes, a temperature $t°$ being maintained. Only one curve, the *rectangular hyperbola*, has ordinate × abscissa constant throughout, and that is the form of the curve AB. Although always approaching the co-ordinates OC, OD, it only meets them at infinity.

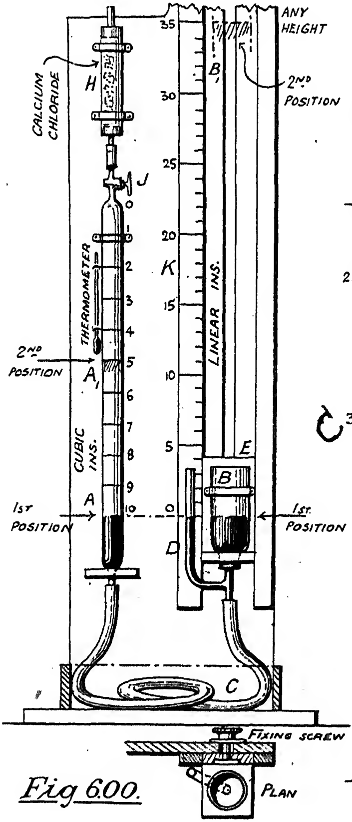


Fig 600.

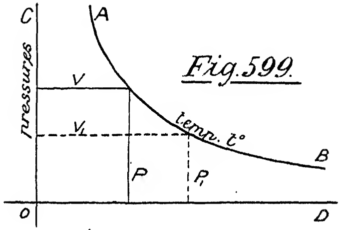


Fig. 599.

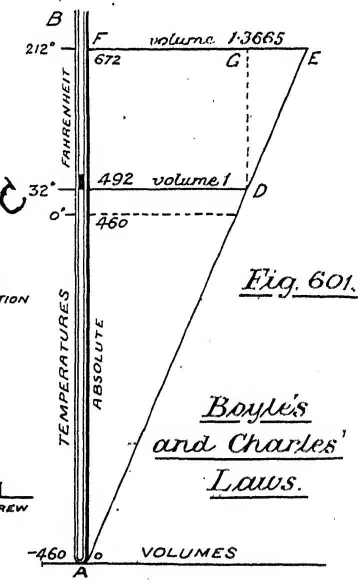


Fig. 601.

Boyle's and Charles' Laws.

Isothermals.—By reason of equality of temperature, AB is also known as the *isothermal of a perfect gas*, that is, of a gas following Boyle's law perfectly. Marriotte's tubes, Fig. 600, prove fairly well the accuracy of this law. A is a closed and B

[illegible] glass [illegible] atmosphere, [illegible]

[illegible] the air supports one atmosphere, and has a volume of [illegible], or $P \times V$ is constant. Intermediate experiments have been obtained and the law more generally proved. The so-called permanent gases are practically perfect, and others become so if measured at a much higher temperature than that of liquefaction.

The second law of gas expansion was discovered by Charles in 1787, published by Dalton in 1801, and by Gay Lussac in 1802, all independently. The last named clearly stated the law, which states that *the increase in volume of a given quantity of gas varies directly as the increase in temperature, if the pressure be constant*; or, if V = the volume at 32°, V_i the increase, V_2 the total volume after increase, and $t°$ the rise in temperature,

$$V_i \propto t°, \quad \text{and} \quad V_2 - V_1 = a t°$$

a being the *coefficient of cubical expansion*. V and a are constants, and $t°$ the only variable;

$$V_2 = V_1 + V_1 a t° = V_1 (1 + a t°)$$

The coefficients of linear expansion for solids, p. 569, vary with the substance, as also their cubical coefficients, being three times the linear ones; but all gases not only expand *regularly*, but each to the same amount, increase of temperature being equal, one coefficient serving for all. Between 32° and 212° the total expansion is .3665 V, or $\frac{.3665}{180}$ = .002036 V, for each degree, figures found by Gay Lussac, expanding the air in an air thermometer, the bulb dipping in heated water, whose temperature was taken by mercury thermometer.

Absolute Zero of Temperature. Let ab, Fig. [illegible], be an air thermometer with an air tight piston c, and let the volume

as be called τ, the temperature [illegible] for volume at 32°, and [illegible] that at 212° [illegible] 1.3665, and the gradual [illegible] straight line [illegible]. Supposing the law [illegible] [illegible] production of [illegible] the temperature, ultimately [illegible] volume will have decreased [illegible] been taken out of the air. [illegible] their supposition enables [illegible] advantages in thermo-volumetric calculation.

To find a, the absolute zero of temperature, we [illegible] similar triangles

$$\frac{ac}{[illegible]} = \frac{100}{[illegible]} \text{ and } ac = [illegible] = [illegible] = 492 \text{ [illegible]}$$

$$\therefore a\text{'s reading} = 492 - 32 = 460° \text{ below zero F.}$$

Any ordinary temperature t° may, then, be made absolute by adding 460, and while t° indicates Fahrenheit readings τ will show absolute readings.

Note that Fig. [illegible] is a graphic statement of Charles' law, as being an isopiestic or line of constant pressure, as [illegible] is a line of constant temperature.

Combination of Boyle's and Charles' Laws. PV is invariable for any particular position on the thermometric scale; but if t° be raised, the value of PV will be raised also. In Fig. 601, if P be kept constant V will vary as τ; so if V increases at the same rate as τ, any series of multiples of V will constantly increase; and as P would be such a multiplier on Fig. 601,

$$PV \propto \tau \quad \text{and} \quad PV = c\tau,$$

which is strictly general, c being a coefficient depending on the gas. (See p. 1128.)

Taking one pound of air at a *temperature of 32°, and at atmospheric pressure*, reckoning in lbs. per sq. ft. and in cubic feet, Regnault found by experiment that

$$PV = 26{,}214 = c\tau, \quad \therefore c = \frac{26{,}214}{32 + 460} = 53.28.$$

For superheated steam $c = 85.5$ (quoted p. 604).

The above formula gives P or V at any temperature, when c is known.

Three States of Matter.—These, the solid, liquid, and gaseous, are well understood, and it is also now admitted that all bodies are capable of existence in each state successively, though not necessarily at the ordinary pressure and temperature. Taking one pound of any substance and applying the specific heat due to its state, its temperature rises one degree, and as the specific heat is approximately regular for each state, practically the whole heat is registered on the thermometer. But in all substances two critical points occur called the points of *fusion* and *evaporation*, and known respectively in the case of water as the 'freezing and boiling points;' at these points additional heat is absorbed merely to do the work of re-arranging the molecules, of fusing or melting on the one hand, and of evaporating on the other hand. Such 'latent' heat is not observable on the thermometer, and must, therefore, be otherwise detected.

Latent Heat *is the quantity of heat units absorbed or given out in changing one pound of a substance from one state to another without altering its temperature.* This phenomenon, first observed by Black about 1757, will now be demonstrated in the case of water, and the units measured.

Latent Heat of Water is that required to melt one pound of ice at 32° F. Provide a vessel with felt-covered sides, similar to that at Fig. 598. Fill it with water of known weight (w) and temperature ($t°$). Take a piece of ice which has begun to melt, wipe dry, weigh (w_1), place in the water, and close the apparatus. When the ice is quite melted, gently stir, and measure the final temperature (T°), which may be a few degrees above 32°. Let L_h = the latent heat of water; then

$$\begin{array}{ccc} \text{Heat lost by water} & = & \text{Heat gained by ice} \\ \text{weight} \times \text{fall of temp.} & = & \text{weight} \times (\text{latent ht.} + \text{rise of temp.}) \\ w\,(t° - T°) & = & w_1\,\{L_h + (T° - 32°)\}. \end{array}$$

Supposing 20 oz. of water at commencement, at 60°, and 2 oz. of ice at, of course, 32°; the final temperature being 45°, then

$$20\,(60 - 45) = 2\,(L_h + 45 - 32),$$

$$\text{and } L_h = \frac{300 - 26}{2} = 137 \text{ units.}$$

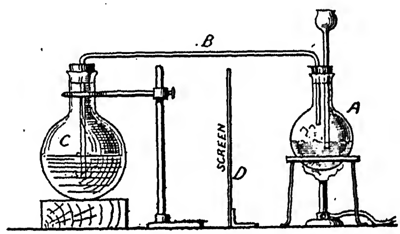
Supposing, further, that one degree in the final
has been gained by radiation from the room, 44
temperature due to mixture,

$$20\,(60 - 44) = 2\,(L_h + 44 - 32)$$

$$L_h = \frac{320 - 24}{2} = \underline{148 \text{ units}}.$$

The correct result should be 144 units, only
careful preliminary radiation experiments.

Latent Heat of Steam.—In Fig. 602 water is
flask A, and steam then passed by tube B to flask C,
into water. The screen D is to prevent radiation



Latent
of Ste

Fig.

and the experiment is continued till the water nearly
Weighing C both before (w_1) and after the
difference is the amount of steam condensed (w). Then

$$\text{Heat lost by steam} = \text{Heat gained by water,}$$
$$w\,\{(212 + L_h) - T^\circ\} = w_1\,(T^\circ - t_1{}^\circ).$$

Suppose the weight of water is 20 oz. at temperature
weight of steam condensed $1\frac{1}{2}$ oz., and the final temperat
a loss of 1° occurring by radiation,

$$1{\cdot}5\,(212 + L_h - 147) = 20\,(147 - 70),$$

$$\therefore\ L_h = \frac{1540 - 97{\cdot}5}{1{\cdot}5} = \underline{961{\cdot}6 \text{ units}}.$$

The exact value is 966 units.

It should be well grasped that latent heat is a kind
heat given to the body *during* the change from solid to

[illegible] of heat is given out [illegible] for every degree, and [illegible] boiling. Water between [illegible] [illegible] for every degree [illegible] [illegible] [illegible]

Fig. [illegible] shows [illegible] of a [illegible] with [illegible] corresponding temperatures. [illegible] of heat required to change both [illegible] and temperature under atmospheric pressure. [illegible] on the diagram [illegible]

Saturation and Boiling Points.—If a boiler be open, the steam is formed under atmospheric pressure, or **14.7 lbs. per sq. in.**, when it [illegible] boils, and its temperature will be 212°. By covering the vessel with a weighted valve, steam is formed at a higher temperature, because under greater pressure. If the water be heated in a partial vacuum, the temperature will be below 212°, because the pressure is reduced. When ebullition first commences, and steam is emitted (see *a*, Fig. 609), the boiling point is reached, and the temperature has a definite value in accordance with the pressure. The steam now forming is in contact with the water, and, being more or less full of watery particles, is called *Wet Saturated Steam*. The latent heat is gradually absorbed, and, when fully taken up [illegible], when all the water has *just* boiled away, the *saturation point* is reached, and we have *Dry Saturated Steam* (see *a*, Fig. 609). Applying heat still further without further supply of water, expansion takes place (approximately) according to Gay Lussac's formula, and the steam is said to be superheated. The boiling and saturation points then, although having the same temperature, by no means represent the same condition. In practice, dry saturated steam is only approximated to by providing domes to boilers, to take the steam as far from the water as possible (see [illegible]).

Besides having specific temperature and pressure, dry saturated steam has a specific volume. In Fig. 609, the volume of steam to that of water is [illegible], called its *relative volume*, [illegible]

the volume of 1 lb. weight is 26·36 cubic ft., termed
volume: and the latter always = *relative volume* × ·016

Def. 1.—**The Saturation Point** *is attained w*
latent heat required for the steam has been

Def. 2.—**The Boiling Point** *occurs when the te*
water overcomes the surrounding pressure.

Def. 3.—**Dry Saturated Steam** *is that which h*
volume, pressure, and temperature, correspon
complete formation.

Def. 4.—**Wet Saturated Steam** *is in process of*
and is in contact with the water.

Def. 5.—**Superheated Steam** *is that which h*
perature raised above formation point.

Def. 6.—**Specific Volume** *is the number of cub*
lb. weight, and SPECIFIC DENSITY *is the num*
in a cubic ft. (*See pp.* 766 *and* 933.)

Dryness Fraction.—If the weight of water par
given volume of wet steam be measured by suitable
the *proportionate wetness* will be shewn when that
divided by the total weight of dry steam and water
while the *proportionate dryness*, or

$$\text{Dryness fraction} = \frac{\text{weight of dry steam}}{\text{weight of steam and water}}$$ (*See Appen.*
pp. 764 *an*

Curves of Saturation Points.—The comp
temperature and pressure of dry saturated steam h
proved by experiment. From − 22° to 32°, Gay-Lussac
apparatus in Fig. 604. Both barometer tubes have va
the mercury, but B has a little water on the surface of the
whose vapour pressure reduces the height of the col
1 in. of mercury represents about ½ lb. per sq. in., the p
therefore known. Various freezing mixtures successively
the blind end of tube B, their temperature being
thermometer.

Fig. 605 was Regnault's apparatus for temperatures
to 122°. As before, barometer A has a perfect vacu
B's vacuum is impaired by vapour rising from water lying

Diagram showing the Relative Volumes and Temperatures of Ice, Water & Steam; together with Heat supplied for conversion.

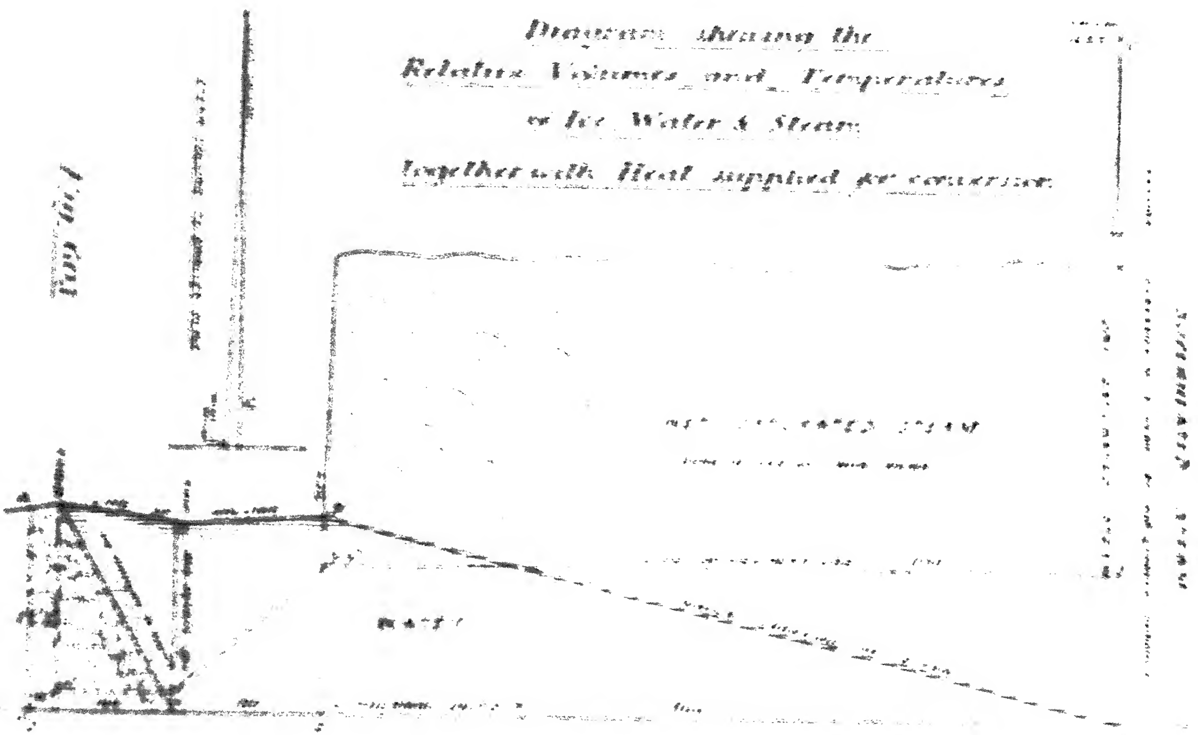
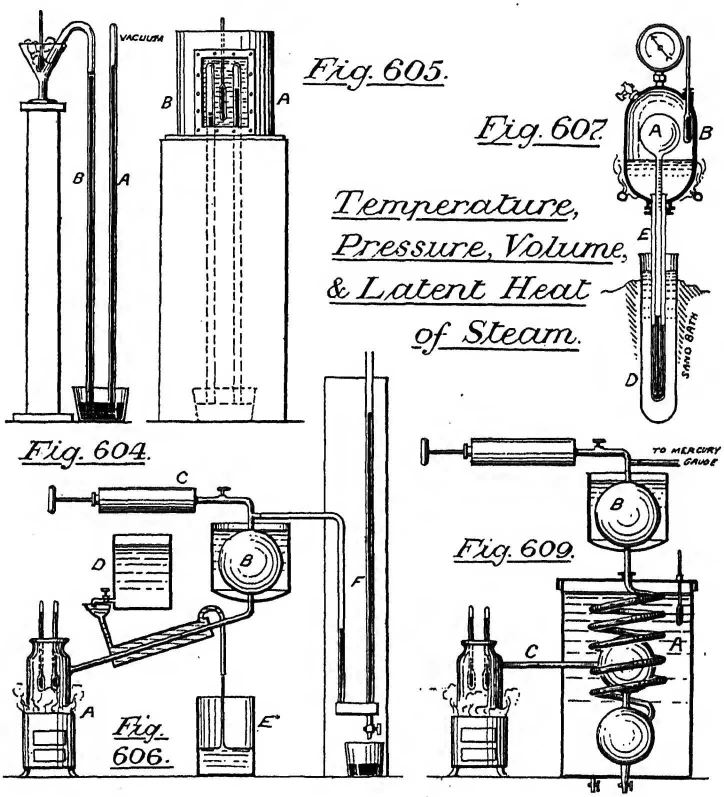


Fig. 601

surface of the mercury. Both tubes are surrounded by heated water, whose temperature is shown by thermometer.

Regnault further found, as in Fig. 606, the temperatures and pressures of saturated steam between 122° and 219°, the experi-



ment having since been carried to 432°. A is a boiler where steam is formed, and B a copper sphere containing an artificial atmosphere, produced by the condensing syringe C. As fast as steam is formed, it is condensed by water passing from D to E round the steam pipe; but this is a practical detail. Essentially the pressures in B and A balance, being measured by the open

[illegible]

[illegible] ... and the water heated, and saturated steam [illegible] equal temperature and equal pressure, [illegible] constant. But when water in the steam evaporates, steam begins to superheat, and its pressure [illegible] the mercury levels separate. Just then the steam is dry, and the temperature, volume of [illegible], and weight of water [illegible] specific volume may be deduced.

Fig. [illegible] shows the results of the above described experiments. *Absolute pressures** are measured on the vertical centre line; to the right are the corresponding specific volumes, and to the left the temperatures Fahr. The former is also a curve of expansion for dry saturated steam, that is, steam *kept always at saturation point.* (*See Appendices I and III, pp.* [illegible], [illegible], *and* [illegible].)

Total Heat of Evaporation *is the quantity required to raise one lb. of water from 32° to a given temperature, and then evaporate it.* The investigation of total heats at various temperatures was successfully pursued by Regnault. Referring to Fig. [illegible], steam was passed through a coiled pipe [illegible] surrounded by water, to which the latent heat was given up, an artificial atmosphere being introduced at p, while thermometers showed temperature of both boiler and tank. The condensation water drawn off at bottom showed weight of steam used. From his results, Regnault deduced the empirical formula

Total Heat per lb. weight T H = [illegible] + [illegible] ($t°$ − 32°),

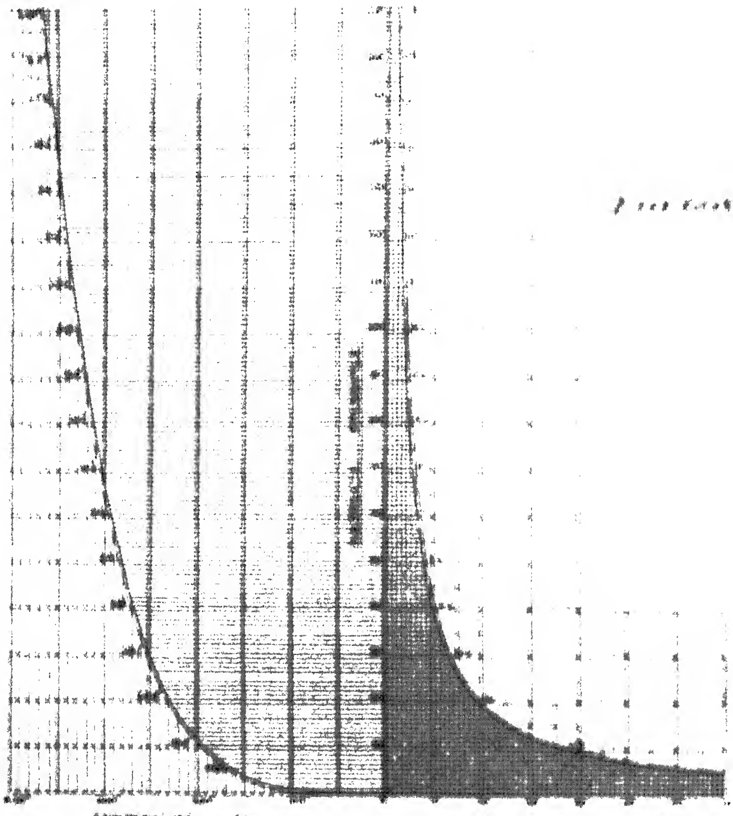
from which, deducting the sensible heat $t°$ − 32°,

L = [illegible]

Formulæ serviceable both above and below 212° if the steam be saturated.

* [illegible]

Pressure, Temperature & Volume of Dry Saturated Steam

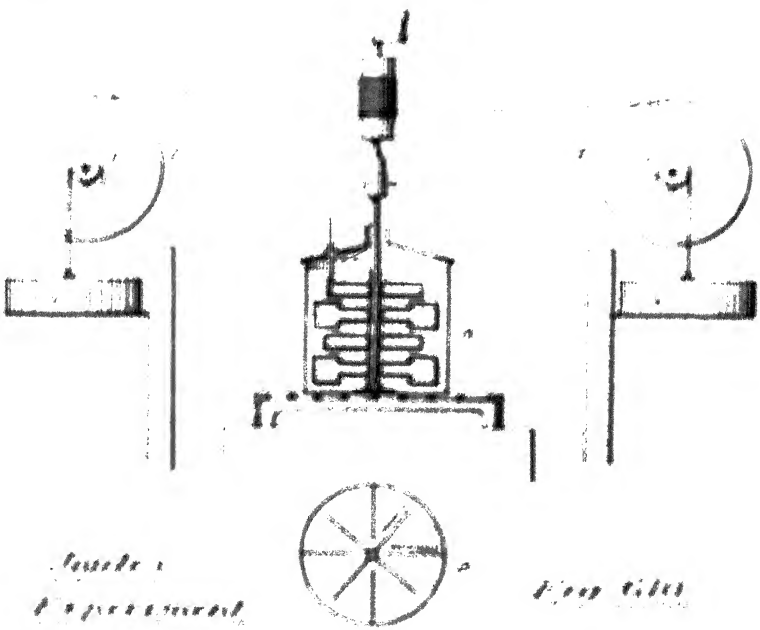


Mixtures of Steam and Water.—We can now calculate the quantity of condensing water required with a given temperature of steam. For convenience, we shall measure from 0° Fahr., and omit the I_a of water.

Example 60. The temperature of condensing water being 60° and that of the exhaust steam 153°, while the condensate remains at a temperature of 120°. Find the weight of condensing water per lb. of exhaust steam. (Hons. Steam E.)

[illegible]

Mechanical Equivalent of Heat. [illegible] ... and Joule ... Mayer ... water by



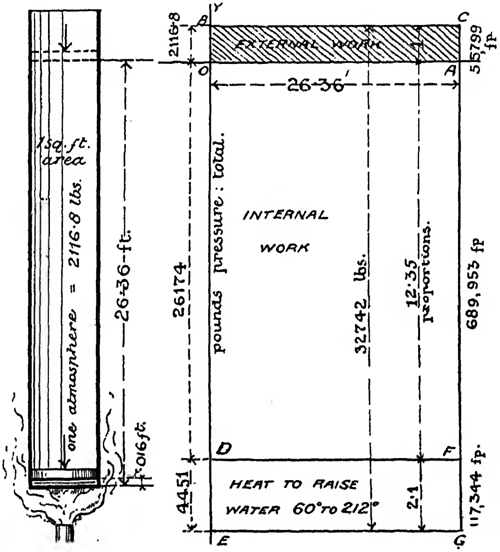
Joule's Experiment

Fig. 619

mere agitation. Adopting the latter method among many others, Joule at the same time measured the work required to raise the temperature. A cylindrical copper vessel (a, Fig. 619) containing water, had diaphragms as at n, through slits in which paddles revolved on the vertical axis c. From roller [illegible] passed light cords to the large pulleys B, B, upon whose axes were smaller pulleys r r

From the circumferences of FF weights GG were hung, being allowed to fall, rotated the paddles and raised th[e tem]perature of the water. By repetition, the temperature [of the] water was raised to a measurable quantity, the work of the weights being simultaneously noted, until the average of [the] experiments gave the 'mechanical equivalent' as 772 foot l[bs.] one British Thermal Unit. We may now state the

First Law of Thermodynamics.—*Heat and mec[hanical] energy are mutually convertible, and Joule's equivalent* (*J*) *rate of exchange.* (*See pp.* 930 *and* 1130.)



Internal & External Work during evaporation. Fig. 61

Internal and External Work during Evaporati[on].—In heating water and evaporating it:

1.—The temperature of the water has been raised.

2.—The water has been changed into steam at the [same] temperature.

3.—The volume of the water and steam has been increa[sed] against external resistance.

* Rowland's later value, 778, is probably more nearly correct.

[illegible]

[illegible]

1 cu. ft. of water weighs 62.5 lbs.

1 lb. of water will stand at $\frac{1}{62.5}$ = .016 ft. high.

Specific volume of 1 lb. of steam at 212° = 26.36 cubic ft.* Heating from 32° to 212° does not materially change the water volume (.016 cu. ft.). But when all the water has become steam, the piston would assume the dotted position ... having risen 26.36 ft. against a pressure of 2116.8 lbs. (14.7 × 144 = 2116.8), thus absorbing 55,799 ft. lbs. of work. By our formula, we know that 966 thermal units, or 966 × 772 = 745,752 foot lbs., have been required, so that 745,752 − 55,799 = 689,953 foot lbs. have been used as *internal* work energy to change the state, and 55,799 foot lbs. have done *external* work against the atmosphere during that change. In addition, there have been required, to raise the temperature of the water, 212 − 32 = 180 thermal units, or 180 × 772 = 138,960 foot lbs. Collecting these results, we have

Sensible Heat = 138,960 foot lbs. to raise water's temperature

Latent Heat = 689,953 foot lbs. to change from water to steam, or as a maximum internal work; = 55,799 foot lbs. to raise piston against atmosphere, or external work.

Total, 884,712 = 1146 × 772 [illegible]

And the three values have the proportions [illegible]

* Specific volume of saturated steam at any other pressure is found from the formula of Rankine or Zeuner, respectively:

$$[illegible]$$

Commencing at O, Fig. 611, draw the co-ordinates OY, OX, for pressure and distance respectively. Measure 26·36 ft. at OA, and 2116·8 lbs. at OB; the rectangle AB then shews external work. Make OD and DE 12·36 and 2·1 times OB respectively; the area OF is the internal work during evaporation, and DG shews the work required to raise the water's temperature from 60° to 212°. Rectangle AB represents the only *useful* effect, the rest being expended on internal changes, and the

$$\underline{\text{Efficiency of the steam}} = \frac{\text{external work}}{\text{total work}} = \frac{55{,}799}{863{,}096} = \underline{\cdot 0646}$$

Let us next examine the case of steam at 160 lbs. pressure (above atmosphere), as in triple-expansion engines.

1 lb. of steam at 174·7 lbs. per sq. in. absolute has a specific volume of	2·5	cub. ft.
Load on piston = 174·7 × 144 . . . . . . =	25,156	lbs.
(1) External work = 25,156 × 2·5 . . . . . =	62,890	ft. lbs.
Temperature of steam . . . . =	370°	F.
Latent heat = {966 − ·7 (370 − 212)} × 772 =	660,369	ft. lbs.
(2) Internal work = (660,369 − 62,890) . . =	597,479	ft. lbs.
(3) Heat to raise water's temperature = (370 − 60) 772 =	239,320	ft. lbs.
And total work = (1) + (2) + (3) . . . =	899,689	ft. lbs.

$$\underline{\text{Efficiency of steam}} = \frac{\text{external work}}{\text{total work}} = \frac{62{,}890}{899{,}189} = \underline{\cdot 07.}$$

Which proves that high pressure steam, *weight for weight* and *without expansion*, is not more economical than low pressure steam.

Specific Heats of a Gas.—As with solids and liquids these are the quantity of heat required to raise the temperature of 1 lb. weight through one degree F. But there are two methods of raising the temperature, the specific heat being a different quantity for each case. Assuming the gas enclosed in a cylinder and covered with a loose piston, we may, while supplying heat, (1) allow the piston to rise freely, or (2) fix it immovably. In we are heating at *constant pressure*, and in (2) at *constant*

[illegible] a larger heat [illegible] the external work [illegible] performed in addition. [illegible] specific heat at constant pressure [illegible]

[illegible] 1 lb. of air at 32° weighs [illegible]

[illegible] standard at = [illegible] = 12·4 feet

under atmospheric pressure and temperature of 32°. Heat to 212°. Then, from Eq. [illegible],

$$\text{Increase of volume} = \cdot 3665 \times 12 \cdot 4 = 4 \cdot 54 \text{ feet},$$

which is also the rise of piston against 2116·8 lbs.

$$\text{External work} = 2116 \cdot 8 \times 4 \cdot 54 = 9610 \cdot 27 \text{ ft.-lbs.}$$
$$\text{Total work} = \text{rise in temp.} \times \text{spec. ht.} = 180 \times \cdot 2375 = 42 \cdot 75 \text{ T.U.}$$
$$= 33{,}003 \text{ ft.-lbs.}$$
$$\text{Internal work} = 33{,}003 - 9610 \cdot 27 = 23{,}392 \cdot 73 \text{ foot pounds},$$
$$\text{or } 30 \cdot 30 \text{ T.U.}$$

But the last figure is the heat required to raise the temperature at constant volume through 180°.

$$\textit{Specific heat at constant volume} = \frac{30 \cdot 30}{180} = \cdot 1683 \text{ T.U.}$$

or, more correctly, may be taken at ·1686, and the ratio of the two specific heats,

$$\gamma = \frac{c_p}{c_v} = \frac{\cdot 2375}{\cdot 1686} = 1 \cdot 408$$

a number we shall require later.

We may also represent the specific heats of a gas in foot-lbs., using symbol K instead of c. Then,

$$K_p = \cdot 2375 \times 772 = \underline{183 \cdot 35 \text{ foot pounds.}}$$
$$K_v = \cdot 1686 \times 772 = \underline{130 \cdot 16 \text{ foot pounds.}}$$

Regnault's Law. *The specific heat of a gas at constant pressure is the same at all temperatures.* This is a most important law, shewing that gases, unlike liquids, expand regularly for regular increments of heat.

Let us heat a gas under a constant pressure P, the being increased from V_1 to V_2 and the temperature ris τ_1 to τ_2 absolute : then

$$\begin{aligned} \text{External work} &= P\,(V_2 - V_1) = c\,(\tau_2 - \tau_1) \\ \text{Total heat expended} &= \text{spec. ht.} \times \text{rise in temp.} \\ &= K_p\,(\tau_2 - \tau_1) \\ \text{and Internal work} &= \text{Total} - \text{External} \\ &= K_p\,(\tau_2 - \tau_1) - c\,(\tau_2 - \tau_1) \end{aligned}$$

But, when a gas is heated at constant volume, only work is done.

$$\therefore\ K_v\,(\tau_2 - \tau_1) = K_p\,(\tau_2 - \tau_1) - c\,(\tau_2 - \tau_1)$$

$$\text{and } \underline{K_p - K_v = c.}$$

Note that internal work is always K_v (final temp. temp.), and may therefore be positive, negative, or not

Specific Heats of Superheated Steam.—By ex $K_p = 370{\cdot}56$ foot lbs., and as steam a few degrees above tion point is a practically perfect gas, K_p will be a regular Further, if we are heating at constant pressure,

$$\text{For steam } PV_s = c_s\,\tau \qquad \text{For air } PV_a = c_a\,\tau$$

$$\text{or, } \frac{V_s}{V_a} = \frac{c_s}{c_a}.$$

Now the ratio of specific volumes $\dfrac{V_a}{V_s} = {\cdot}622.$

$$\therefore\ \frac{c_s}{c_a} = \frac{1}{{\cdot}622} \text{ and } c_s = \frac{c_a}{{\cdot}622} = \frac{53{\cdot}2}{{\cdot}622} = \underline{85{\cdot}5.}$$

Then $K_p - K_v = 85{\cdot}5$ and $K_v = 370{\cdot}56 - 85{\cdot}5 = 285{\cdot}06$

$$\text{Finally } \gamma = \frac{K_p}{K_v} = \frac{370{\cdot}56}{285{\cdot}06} = \underline{1{\cdot}3.}$$

Expansion Curves and their Areas.—The hy co-ordinating Boyle's law, has been shewn at Fig. 599. other expansion curve, as these are called, has the formul the exponent n changing with the substance. Now the *area*, Fig. 612, shews the work done during expansion, as

be actually measured (see Fig. 325); but as these curves have definite formulæ, it is easier to use algebraic methods. Then,

Area of curve having formula PV = C is $PV \times \log_e \frac{V_2}{V_1}$

and as $PV = c\tau$, and $\frac{V_2}{V_1}$ = the ratio of expansion r.

$$\underline{\text{Area} = c\tau \log_e r.}$$ (*See p.* 1131.)

(Use hyperbolic logarithms, and see Fig. 612)

Area of curve having formula $PV^n = C$ is $\frac{P_1V_1 - P_2V_2}{n-1}$

Isothermals and Adiabatics.—If a gas expand, and advance a piston against a resistance, it does work requiring expenditure of heat. Such heat being abstracted from the gas, the temperature of the latter falls; but if heat be supplied just as fast as it is abstracted, viz. equal to the work done, the temperature will remain constant, the expansion be according to Boyle (PV = C), and the curve be called an *isothermal.*

If no heat be supplied, the pressure-volume curve will fall below the hyperbola, as in Fig. 613, according to the formula $PV^n = C$, and be then termed an *adiabatic.* Similarly, in compressing, the adiabatic will rise above the isothermal, because the gas becomes heated by work done *upon* it (Fig. 614). (*See Appendix II., p.* 879.)

Adiabatic Exponent.—The value of n will now be found for the adiabatic.

$$\text{Area of curve} = \frac{P_1V_1 - P_2V_2}{n-1} = \frac{c}{n-1}(\tau_1 - \tau_2) = \text{External work.}$$

$$\text{Total work} = \text{Internal work} + \text{External work}$$

$$= K_v(\tau_2 - \tau_1) + \frac{c}{n-1}(\tau_1 - \tau_2)$$

$$= (\tau_2 - \tau_1)\left(K_v + \frac{K_v - K_p}{n-1}\right)^*$$

$$= (\tau_2 - \tau_1)\left(\frac{K_v(n-1) + K_v - K_p}{n-1}\right) = (\tau_2 - \tau_1)\left(\frac{nK_v - K_p}{n-1}\right)$$

* Notice change of sign in two places in order to balance.

Expansion Curves.

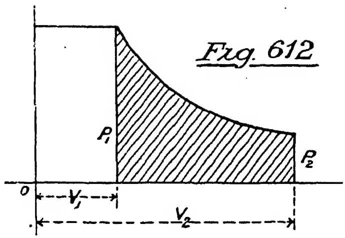


Fig. 612

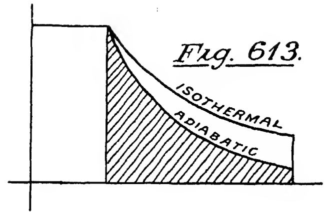


Fig. 613.

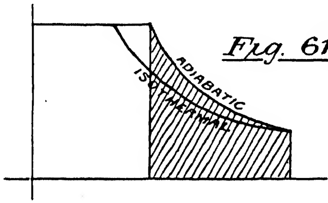


Fig. 614.

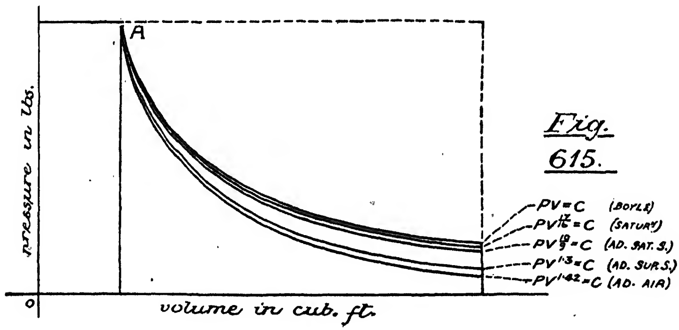


Fig. 615.

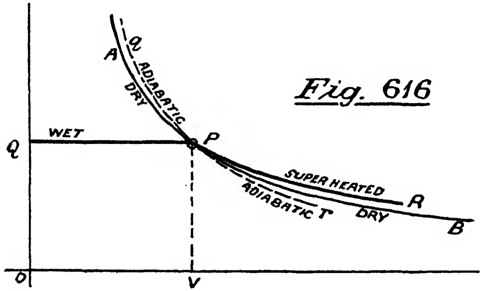


Fig. 616

But in the adiabatic no heat is communicated to the gas, and none taken away.

$$\text{Total heat supplied} = [illegible] \left(\frac{n K_v}{n} - \frac{K_p}{\gamma} \right) [illegible]$$

that is, one of the bracketed quantities is nothing. But [illegible] is tangible,

$$\frac{n K_v}{n} - \frac{K_p}{\gamma} = 0 \qquad n K_v - K_p = 0$$

$$\text{and } n = \frac{K_p}{K_v} = \gamma$$

PV^γ is the general equation for the adiabatic.

In adiabatic expansion, external work is done at the expense of the internal heat of the gas. (See *Appendix II, p.* [illegible].)

Comparison of Temperatures in Adiabatic Expansion. Suppose that by adiabatic expansion the temperature falls from τ_1 to τ_2.

$$P_1V_1^\gamma = P_2V_2^\gamma \quad \text{and } P_1V_1(V_1)^{\gamma-1} = P_2V_2(V_2)^{\gamma-1}$$

$$P_1V_1 = P_2V_2\left(\frac{V_2}{V_1}\right)^{\gamma-1} \text{ or } \tau_1 = \tau_2\left(\frac{V_2}{V_1}\right)^{\gamma-1}$$

$$\tau_1 = \tau_2\left(\frac{r}{1}\right)^{\gamma-1} \text{ or } \tau_1 = \tau_2\left(\frac{r}{1}\right)^{[illegible]} \text{ for air}$$

Various Expansion Curves, as represented by their formulæ, may now be collected as follows:

For *isothermal expansion* of a perfect gas,

$PV = C$ (a true rectangular hyperbola).

For *adiabatic expansion* of a perfect gas,

$PV^\gamma = C$ (γ = 1·408 for air, 1·3 for superheated steam).

For *expansion of dry saturated steam*, without becoming wet or superheated, being the "curve of saturation points," Fig. [illegible],

$pV^{17/16} = C = 475$ (Rankine's curve).

$(p + [illegible])(V - [illegible]) = C = [illegible]$ ([illegible]'s curve).

Both founded on Regnault's experiments.

For *adiabatic* expansion of *saturated steam*,

$$pV^{1.135} = C \quad \text{(Zeuner's curve)}$$

$$pV^{\frac{10}{9}} = C \quad \text{(Rankine's curve)} \quad \text{(See Fig. [illegible], p. [illegible].)}$$

For *adiabatic* expansion of *superheated steam*,

$$pV^{1.3} = C$$

The above adiabatics represent the expansion of steam in a cylinder under good conditions. All starting from the same point, A, Fig. 616, the hyperbolic curve is highest, then the saturation curve, the adiabatics for saturated and superheated steam respectively, and lastly the adiabatic for air.

Isothermals of Saturated Steam or other Vapours. In Fig. 616, A B is the saturation curve, and P a point showing *dry saturated steam* at pressure P, volume V, and corresponding temperature. If V be decreased by compression, temperature being constant, some steam liquefies, P is kept constant, and the compression curve is P Q, the steam becoming *wet*. If, again, V be increased at constant temperature, the steam becomes superheated, and expands along P R, rising above the saturation curve P B, which is a curve of lowering temperature. Q P R is sometimes called the expansion curve of *dry* saturated steam—an incorrect description, for the steam is only dry at one point P. The adiabatic $q\,r$ has the formula $pV^{\frac{10}{9}} = C$, or $pV^{1.135} = C$. (See p. 766.)

Cycle of Operations. If a working gas be passed through a series of heat changes, and ultimately returned to its original condition, the changes constitute a *cycle*, and *external work* has been done equal to the *heat expended*, because the gas, reverting to its original state, will have returned the *internal work* first absorbed. The indicator card represents a particular cycle.

Carnot's Reversible Cycle or Perfect Heat Engine is the most perfect example of an engine cycle. It should be understood that the object of making changes upon a gas is to obtain external work from heat; and though Carnot's engine is unattainable in practice, yet its perfection should be approached as closely as possible by practical engines. All engines, Carnot's included, receive heat energy from some *hot body*; during expansion of the working substance, give external work to moving

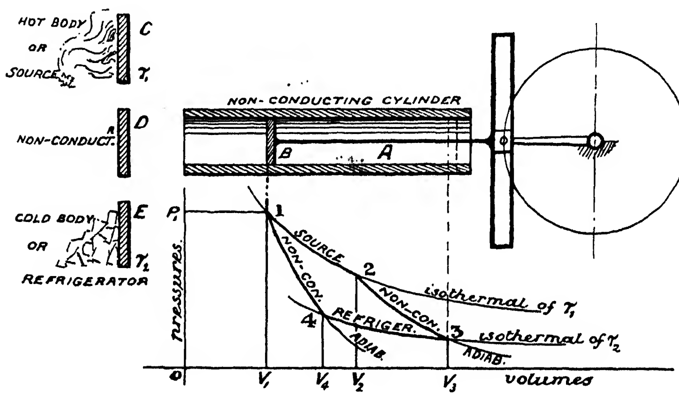
mechanism; and, finally, reject a smaller quantity of heat into some *cold body*. In the steam engine these 'bodies' are the boiler and condenser respectively. We shall see that the efficiency of the engine does not depend on the working substance, if a reversible cycle be adopted, but only on the difference of temperatures between which the substance is utilised. A perfect heat engine should have the following qualifications:—

1. The heat must be received at the temperature of the hot body.
2. The heat must be rejected at the temperature of the cold body.
3. The cycle must be reversible.

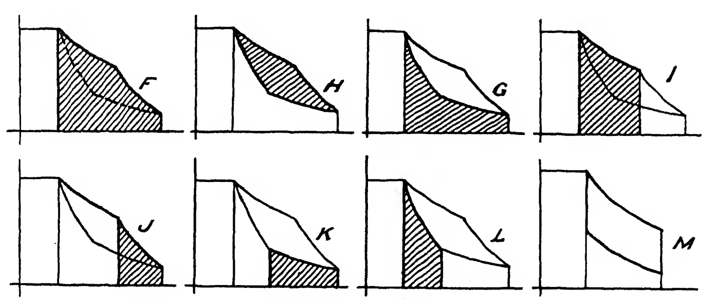
For perfect working, it is clear that *all* heat represented by drop of temperature between the hot and cold bodies should be delivered to the engine as work. But if there be a fall of temperature between hot body and engine, or between engine and cold body, some heat will be lost on the way which does not reach the engine. Hence the reason for (1) and (2). We may explain (3) similarly, first premising that by *direct action* we mean the transformation of heat into work by abstraction of heat from hot body; *reversed action* being obtained by turning the engine backward, giving all the work back to the hot body. In a perfect engine, the work given by the gas during one direct cycle must equal the heat returned during one reversed cycle, which is to say, that *all* the 'available' heat must be transformed into work.

Carnot's cycle fulfils these three conditions, and none other can have a higher efficiency, as we shall prove. Fig. 617 is the ideal engine, having a non-conducting cylinder A, and piston B, the latter connected to suitable working mechanism. C is the hot body, E the cold body, and D a non-conducting cylinder-cover; and the underlying diagram indicates the changes we are now to follow. *First operation:* Commencing with a portion of gas behind the piston, at temperature τ_1 (that of the hot body), pressure P_1, and volume V_1, we allow this to expand at constant temperature while doing work. Placing the left end of the cylinder on the hot body, the expansion curve is the isothermal

1 2. *Second operation:* The expansion is continued, without supply of heat, by placing the cylinder on the non-conducting cover; and the adiabatic curve 2 3 is traced, the temperature falling from τ_1 to τ_2, on account of work done by the gas. *Third operation:* Compressing the gas at constant temperature τ_2, we place the



Carnot's Engine Fig. 617.



cylinder on the cold body, to receive such heat as must be rejected; and the curve obtained is the isothermal 3 4. *Fourth operation:* Finally, place the cylinder on the non-conducting plate and compress along the adiabatic 4 1; the substance is then returned to its original condition and temperature τ_1.

During these operations the work done *by* the gas is shewn by diagram F, and that *on* the gas by diagram G, their difference

H being the effective work given to the engine. Reckoning the heat used, we have:

From 1 to 2 (r_1). *Heat expended*, being work area I,
$= P_1 V_1 \log_e r_1 = c\tau_1 \log_e r_1$.

From 2 to 3 (r_2). *No heat expended*, external work, J, being done by abstraction of heat from the gas.

From 3 to 4 (r_3). *Heat rejected*, as at K,
$= P_3 V_3 \log_e r_3 = c\tau_2 \log_e r_3$.

From 4 to 1 (r_4). *No heat rejected*, external work, at L, producing internal work on the gas.

We have previously found (p. 607) the comparison of temperature in terms of r, during adiabatic expansion or compression:

$$\tau_2 = \tau_1 \left(\frac{1}{r}\right)^{\gamma - 1}$$

from which may be deduced:

$$r = \left(\frac{\tau_1}{\tau_2}\right)^{\frac{1}{\gamma - 1}}$$

Referring to Fig. 617, expansion from 2 to 3 and compression from 4 to 1 are between the same temperatures, so the *ratio of adiabatic expansion equals that of adiabatic compression*: $r_2 = r_4$

And as, $\dfrac{V_3}{V_2} = \dfrac{V_4}{V_1}$ $\qquad V_1 V_3 = V_2 V_4 \qquad$ and $\dfrac{V_2}{V_1} = \dfrac{V_3}{V_4}$

Or the *ratio of isothermal expansion equals that of isothermal compression*: $r_1 = r_3 = r$, say.

Resuming; when the cycle is complete no internal work has been done—all is external work:

$\therefore$ External work = Heat expended − Heat rejected
$= c\tau_1 \log_e r_1 - c\tau_2 \log_e r_3 = (\tau_1 - \tau_2)\, c \log_e r$.

Efficiency of Engine $= \dfrac{\text{Work done}}{\text{Heat expended}}$

$$= \frac{(\tau_1 - \tau_2)(c \log_e r)}{\tau_1 (c \log_e r)} = \frac{\tau_1 - \tau_2}{\tau_1}$$

(*See pp.* 768, 883, 887, 934, *and* 966.)

It will be easily seen that for the highest efficiency, τ_2 must be nothing, or the condenser must have a temperature of 'absolute

zero,' a condition practically unattainable, and *all the heat in the working substance can never be utilised.* The energy obtainable is only that between the *available temperatures,* and the difference of τ_1 and τ_2 should therefore be as large as is practically possible.

Reversed Action occurs, as previously suggested, when expansion takes place along 1 4, 4 3, and compression along 3 2, 2 1, the operations being entirely the reverse of those just considered. External work is done *on* instead of *by* the gas, and heat is *taken from the cold body and rejected into the hot body.* No better practical example of a reversed cycle can be given than that of an air-compressing engine as at Fig. 562, p. 546.

Let it be possible to have an engine (No. 2) of equal power but higher efficiency than Carnot's (No. 1); and let No. 2 drive No. 1 in reverse order. Then No. 2, taking its heat from the hot body and rejecting into the cold body, and giving all its external work towards driving No. 1, the latter is thus made to take heat from the cold body, which, together with the work received, it delivers into the hot body. No external work being left over, the contrivance is self-acting.

Let H_2 be the heat taken from the hot body by No. 2, and h_2 that rejected into the cold body; H_1 the heat rejected into the hot body by No. 1, and h_1 that taken from the cold body. Power being equal,

$$\text{(Reversed) } H_1 - h_1 = H_2 - h_2 \text{ (Direct)} \quad . \; . \; . \quad (a)$$

Efficiency of No. 1 $= \dfrac{H_1 - h_1}{H_1}$ Efficiency of No. 2 $= \dfrac{H_2 - h_2}{H_2}$

But $\dfrac{H_2 - h_2}{H_2}$ is to be greater than $\dfrac{H_1 - h_1}{H_1}$

And, by (a), the numerators are equal,

$\therefore$ H_2 must be less than H_1. (*See pp.* 770, 773, 883, *and* 1132.)

The heat taken from is therefore less than that given to the hot body, and by a *self-acting process* heat is being taken from the cold and delivered to the hot body, which is impossible by the

Second Law of Thermodynamics.—*Heat cannot pass from a cold to a hot body without external aid.* This is the result of experience, the tendency being always to equalisation

of temperature by heat passage from the hot to the cold body [illegible] we conclude that no engine can have a higher efficiency than [illegible].

While true that the efficiency of a reversible cycle is independent of the substance, there remain practical differences inseparable from the latter. Thus air being an indifferent heat conductor, large surfaces must be presented to allow the heat changes to be made rapidly, and the apparatus becomes unwieldy for high powers. Similarly the nature of the substance in the steam engine prevents its complete compression after condensation, an essential in the perfect engine, and reheating must occur in the boiler. Steam engines have also lower efficiency than has the perfect cycle, for several reasons, included in the following list.

Causes of Energy Loss in the Steam Engine, reducing its efficiency below that of an ideal reversible cycle.

1. Steam is not supplied at the temperature of the hot body, or furnace. This is the greatest loss.

2. Steam is not rejected at the condenser temperature and pressure, but falls regarding both when leaving the cylinder, called incomplete expansion. Further, the temperature of rejection in condenser is higher than t_2 would imply.

3. Steam should be compressed adiabatically from condenser temperature to *furnace* temperature. A small portion is sometimes compressed nearly to *boiler* temperature, but much the major portion has to be reheated in boiler by extra heat supply, when condensed water is returned by feed pump.

4. If the condensed water be not returned to boiler, fresh feed water must be raised from 60° to boiler temperature instead of being originally at temperature of steam.

5. Initial condensation (*page* 614) causes waste throughout the stroke, which is only partly recompensed by re-evaporation. The expansion is not, therefore, adiabatic, as it should be.

6. Clearance (*page* 616) in cylinder being unavoidable, must be filled with steam at every stroke, which does no work during "full pressure" period.

7. The boiler "primes" more or less, that is, sends water particles along with the steam, which pass to the condenser without doing work, or, still worse, abstract heat from the cylinder steam in their attempt to evaporate.

8. The range of working temperature is small in comparison with the temperatures themselves : τ_1 being fixed to prevent burning of cylinder oils and packing, and τ_2 by the cold well temperature.

9. Heat is lost by radiation.

10. The substance is lost by leakage, taking heat with it.

11. Wherever imperfectly-resisted expansion occurs, reversibility is impaired : *e.g.*, the 'drop' into receiver in a compound engine.

12. Various small losses, shewn on indicator diagram : *e.g.*, wire drawing, &c.

13. Work is lost in (*a*) the 'solid' friction of the engine parts, (*b*) the fluid friction of the passing steam. (*See App. III., p.* 935.)

Initial Condensation and Re-evaporation.—When hot saturated steam enters a cylinder cooled to exhaust temperature, an '*initial condensation*' occurs, which is not immediately apparent on the pressure diagrams. After cut-off, further condensation lowers the expansion curve, as shewn dotted at A B, Fig. 618. But cylinder and steam becoming more equal in temperature, the latent heat, liberated during liquefaction, is permitted to raise the curve, as at B C, by causing a certain *re-evaporation.* The first loss is, however, very great, and by no means made up by the second gain, so there is always a quantity of water rejected at release, some of which evaporates during exhaust and creates a back pressure. These losses may be mitigated (1) by applying clothing in quick running engines, and thus securing approximate adiabatic expansion, (2) by adopting a steam jacket for engines of a slower type, where there is time for the heat to be taken up, or (3) by superheating the steam before admission, and partially removing the first cause. The jacket both assists re-evaporation at an earlier, and consequently more available, portion of the stroke, and prevents to some extent initial condensation : the experimental gain being stated at from 10 to 20 per cent. Liquefaction in the jacket is not so detrimental, but in the cylinder the water acts as a conductor from the steam to the metal. Live steam should always be used for the jacket, and efficient drainage applied.

Theory of Compounding.—Another way of decreasing liquefaction is to divide the work among 2, 3, or 4 cylinders; and, if great differences of temperature be employed, no other course is possible. Thus we arrive at the Compound, Triple, or Quad-

[illegible]

Fig. 618

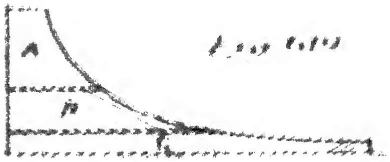


Fig. 619

Fig. 620

Expansion in the Cylinder. Assuming steam to follow Boyle's law, Fig. 620 is a simple diagram of work done. The steam being admitted during, say, a quarter stroke, and the supply cut off, the rest of the stroke is completed by expansion. From A to B there would be *full* steam, and from B to C the pressure would approximately fall along the isothermal and hyperbola B C. Then area O A B D shows work done by the full steam, and D B C F the additional work during expansion.

Construction of Hyperbola. To draw the curve, join O B, produce the crossing point x horizontally to c, which is a point in the curve. Other points being found similarly, between B and C, by projecting from the ends of radial lines, the curve is traced through the crossing points.

* More correctly, 2, 3, or 4 stage compounds.

Clearance. Supposing a certain clearance [illegible] face and piston at end of stroke, to be filled before the piston moves. Representing this by [illegible], in terms of the stroke [illegible], the steam now expands from [illegible] to [illegible], and the hyperbola must be drawn from the new origin [illegible], thus raising the curve [illegible] as dotted, and the rate of expansion will have changed from

$$[illegible] \text{ to } [illegible] \quad \text{or from } [illegible] \text{ to } [illegible].$$

The Steam Engine Indicator is a well-known apparatus (first invented by Watt, and much improved by McNaught and Richards) for the purpose of automatically describing the pressure-stroke diagram just considered. Fig. 6__ represents a 'Tabor' indicator, one of the present varieties of the instrument. A is a small cylinder containing a liner, in which a piston, B, slides freely. Steam being admitted under B, pencil P is raised by the connections, [illegible] being a rocking fulcrum. P would describe an arc but for the slot S, which compensates the curvature and compels the pencil to move in a vertical straight line, its displacement indicating rise or fall of *steam pressure*. The drum D, provided with paper for the diagram, being rotated on stud [illegible] by the cord H (attached to the moving engine) in one direction, and by the clock spring M in the other direction, represents the stroke of the engine piston. Both actions occurring at once, a diagram like Fig. 62_ is obtained. Certain deviations, however, occur, which we shall afterwards discuss.

Figure __ represents the indicator gear usually adopted. It is there applied to a horizontal engine, but may be modified to suit other forms. Lever L vibrates with the crosshead, and carries on its axis the 'Brumbo' pulley [illegible] for decreasing the stroke of the cord to suit that of the indicator drum. The indicator is connected to each end of the cylinder by a pipe provided with stop-cocks at P and Q, and an indicator cock at R. The latter is seen in section at U and V, having a three-way passage to admit the steam (1) to the indicator and out to the air for blowing through, (2) to the indicator only, or (3) the cock may be closed. To avoid clearance in pipe PQ it is better to use two indicators, fixed at P and Q. Notice also the spring W, of which several different

The "Tabor" Steam-Engine Indicator.

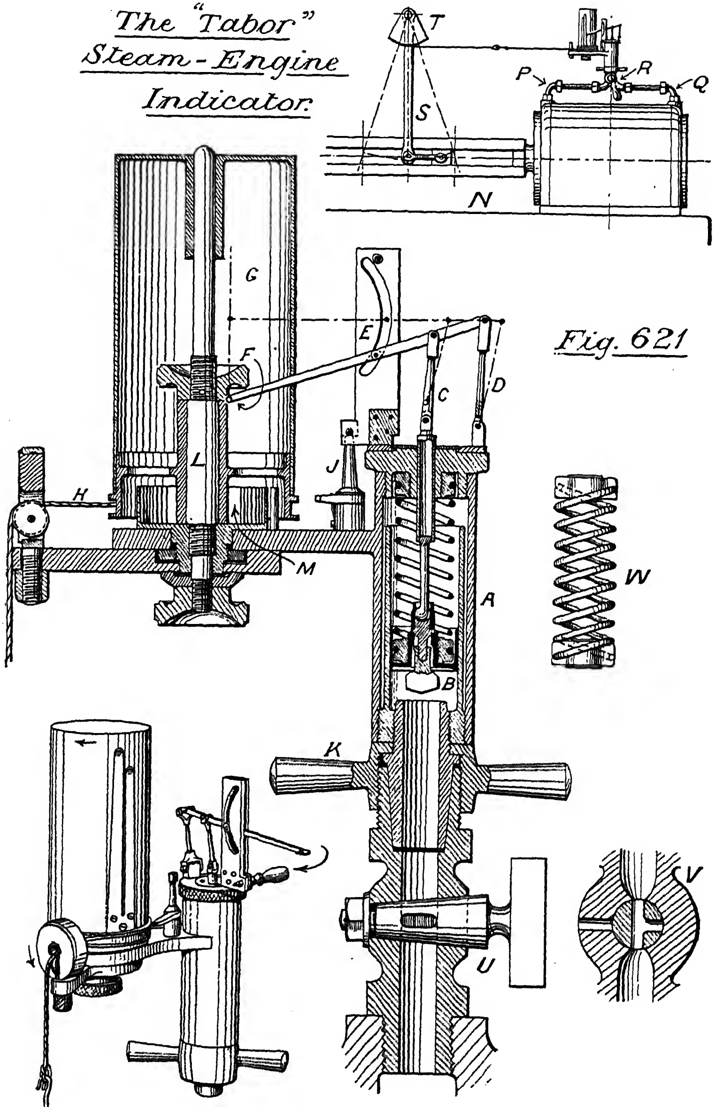


Fig. 621

strengths are provided, so as to indicate pressure to any convenient scale: two wires are coiled in the same direction, but start from opposite sides of the base.

To use the indicator, first let the engine rotate uniformly, then connect the cord. Open cock F and turn E to blow through. Open indicator to atmosphere and let pencil describe atmospheric line; then connecting indicator to steam, bring the pencil gently round and describe the diagram. In like manner also with the cock E2, after which the paper may be removed. The 'pencil' is usually brass wire, and the paper that known as 'metallic.' (*See pp.* 891 *and* 1134.)

Topography of Indicator Diagram. Taking a condensing engine, *i.e.*, one which exhausts into a vacuum, and has additional pressure due to the atmosphere on the forward side of the piston, as in Fig. *622*: O A is the vacuum or *zero* line, and B C the atmospheric line of 15 lbs. absolute; D A is the stroke, and O D the clearance (valve passages and clearance proper) in terms of D A.

The clearance space first fills, and the pressure rises to F. Then the piston moves to G, where steam is cut off, expansion takes place between G and H, release to exhaust at H, pressure falling only to J while the piston returns, because we cannot entirely eliminate vapour pressure in the condenser (back pressure) shewn by A J. Exhaust being fully open between J and K, a horizontal line is drawn up to compression point K, and the remaining steam compressed to L, where it is met by incoming fresh steam, due to the opening (lead) of the valve before commencement of stroke, and the pressure once more rises to F.

Deviations from the Normal Diagram are shewn in Fig. 623. *Wire drawing at cut-off* is indicated at K, the full steam line falling on account of narrow ports or throttling by the slide valve. H shews *late admission*, the piston travelling some distance before full pressure is felt, due to want of lead on the valve, which should open *before* the end of stroke. *Late release* and *excessive clearance* are seen at L, and a *leaky piston* would cause diagram F, the pressures on each side of piston tending to equalise. A *leaky slide valve*, as at G, would raise the expansion curve at the expense of fresh steam, and *initial condensation*, H, is sometimes detected, by drawing the hyperbola. *Too much or too*

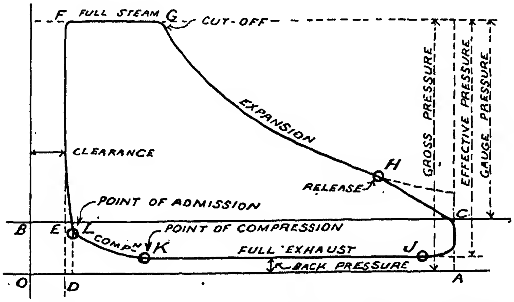
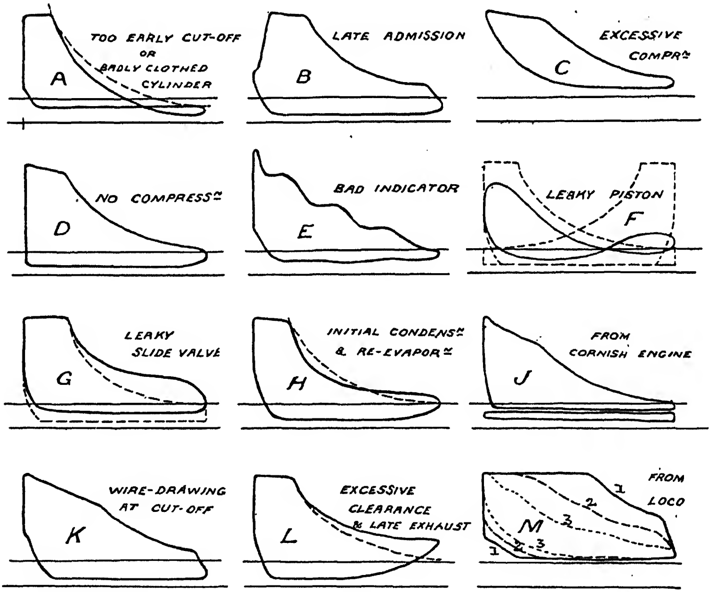


Fig. 622.

Topography of the Indicator Diagram



Indicator Diagram *Fig. 623.*

little compression would give diagrams C or D respectively, and a *shaky diagram*, like E, would be produced by an indicator with too light a spring or too heavy a piston. Diagram A shews

serious cylinder condensation. The upper and lower diagrams at J are from the top and bottom of the piston respectively in a Cornish single-acting pumping engine: and M shews the varying diagrams obtained from a locomotive, (1) when starting, next (2), and lastly (3), as the valve gear is linked up from the reversing lever.

From the Indicator diagram we may therefore deduce:

1. The points of admission, cut off, release, compression, &c.
2. Comparison of cylinder with boiler pressure.
3. The wire drawing in steam and exhaust passages.
4. The back pressure.
5. The condensation, re-evaporation, and relative dryness.†
6. The indicated horse power from the diagram area.

Calculation of Indicated Horse Power, or that shewn upon the indicator diagram, and representing the work given to the piston by the steam or gas.* Three pairs of diagrams, in Fig. 624, are taken from the respective cylinders of a triple-expansion engine; and are copied from the Hons. Engineering Exam. 1887. The *mean effective pressure per square inch* (p) must first be found, so the diagrams are divided into 10 parts by equidistant vertical lines. Knowing the scale of the indicator spring, the pressure may be measured at the middle of each division, *within* the enclosed curve; these figures representing the *effective* pressures. Notice that at A and F, Fig. 623, the loop encloses *minus* effective pressure; every measurement must there be treated as minus, and only added to the other *plus* measurements algebraically. Adding the 10 measured parts, and dividing by 10 gives mean effective pressure for each diagram; the mean of the pair being then found by adding them and dividing by 2.

Multiplying (p) by piston area (a) gives *total mean pressure*, and this again by stroke in feet (L) gives work in foot pounds per stroke. Further multiplying by number of strokes per minute (N) gives work per minute, and the whole divided by 33,000, or one

* Brake horse power is found by dynamometer, as at p. 575, and mechanical efficiency of engine $= \frac{\text{B.H.P.}}{\text{I.H.P.}}$

† *See Appendices I. and II., pp.* 764 *and* 818.

horse power per minute, will represent the indicated horse power of the engine, the formula becoming

$$\text{Indicated horse power (per min.)} = \frac{p \mathrm{L} a \mathrm{N}}{33{,}000}$$

Taking the high pressure cylinder in Fig. 624, the addition of the pressures on the left diagram, 73, 103, &c. = 679·5, and the mean pressure = 67·95. The right diagram similarly has a mean pressure of 59·5, the final mean pressure becoming 67·95 + 59·5 ÷ 2 = 63·72. Area of cylinder is 10 × 10 × 22 ÷ 7 = 314·16, stroke is 3 feet, and number per minute 63 × 2 = 126. We have then:

$$\text{I. H. P. in H. P. cylinder} = \frac{63{\cdot}72 \times 3 \times 314{\cdot}16 \times 126}{33{,}000} = \underline{229{\cdot}3.}$$

In the intermediate cylinder, mean pressure on the left is 23·9 and that on the right 22, the final mean being 22·95. Area of piston = 855·3; stroke and revolutions as before.

$$\text{I. H. P. in I. P. cylinder} = \frac{22{\cdot}95 \times 3 \times 855{\cdot}3 \times 126}{33{,}000} = \underline{224{\cdot}44.}$$

Mean pressures on left and right respectively in low pressure cylinder are 9·5 and 7·65, and the mean of these is 8·57. Then,

$$\text{I. H. P. in L. P. cylinder} = \frac{8{\cdot}57 \times 3 \times 2290{\cdot}22 \times 126}{33{,}000} = \underline{224{\cdot}41}$$

Advantages of Single, Double, and Triple Stage Expansion.—The advantage of expanding steam in a single cylinder, instead of using full pressure to the end of the stroke, was demonstrated by Watt in 1782, and can be understood from Fig. 625. E G is the stroke, and F N that portion during which full steam is used, the rest of the stroke, N G, being completed by the pressure of the expanding steam. From what we know of the work diagram, SBCU will shew work performed by the 'full' steam, without condensation, UCKT that by expansion without condensation, and HSTJ that produced by the use of a condenser Not only then do we obtain additional work by condensing, but we are also enabled with the assistance of high pressure to introduce an earlier cut off and higher rate of expansion: thus using a less weight of live steam.

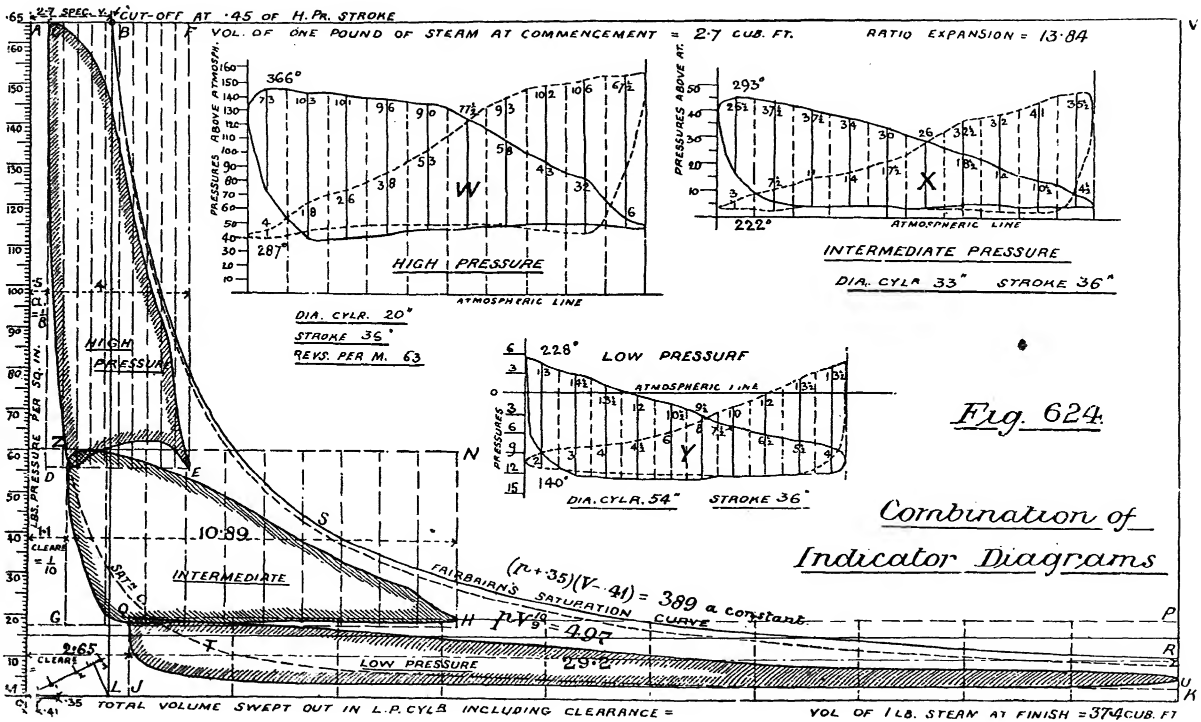


Fig. 624.

Combination of Indicator Diagrams

Though very high rates of expansion are theoretically possible in a single cylinder, the practical advantages of such [illegible] are not realized, for very high rates necessitate very high initial pressure, and a great difference of temperature between live and exhaust steam. The former entering a more or less cold cylinder, considerable initial condensation results, which largely neutralizes the advantage of increased expansion. [illegible] is best avoided by allowing the steam to successively expand through one, two, three, or even four cylinders, thus introducing the so-called compound, triple, and quadruple forms of engines. Referring now to Fig. 624, the fall of temperature in the high pressure cylinder is from 366° to 287° or 79°, that in the intermediate cylinder 71°, and that in the low pressure cylinder 88°, a fairly equal division of the total fall, which, if the work were performed in one cylinder, would be no less than 238°. The total ratio of expansion is nearly 14 : 1, an amount impossible in one or even two cylinders, because of the serious loss from initial condensation occurring throughout the stroke; and it is in this, the diminishing of the effects of initial condensation with high grades of expansion, that the advantage of compounding is most apparent.

One other advantage of dividing the work is that two or three cranks are then employed, set mutually at angles of 90° or 120° respectively, causing a more equable turning effort, as will be more fully demonstrated later, avoiding dead centres. Even before the practical introduction of compounding, double-cylinder engines were found necessary where frequent reversal was required, as in locomotive and marine engines. Each piston should, however, give to its crank, as nearly as possible, the same amount of work as any one of its fellows, a requirement of greater importance than equal distribution of temperature fall. Both points have been well met in Fig. 624, for the total work is divided as 119, 114, 114, while the corresponding temperature drops are 79°, 71°, and 88°. (*See p.* [illegible].)

Combination of Indicator Diagrams in Compound and Triple Engines. Diagrams w, x, and y, in Fig. 624, are just as received from the indicator, where for practical convenience different pound scales are employed; the lengths of the diagrams are also made for our convenience of taking. Now

primarily the base line should be volumetric, so in representing these diagrams to the same scale, the bases must be altered to suit the volume of each cylinder respectively, and one pressure scale be used throughout; we shall then see at a glance the comparative work performed in each cylinder, and shall further be able to judge how nearly the total diagram corresponds with what should take place were the whole expansion to occur in one cylinder under theoretically good conditions.

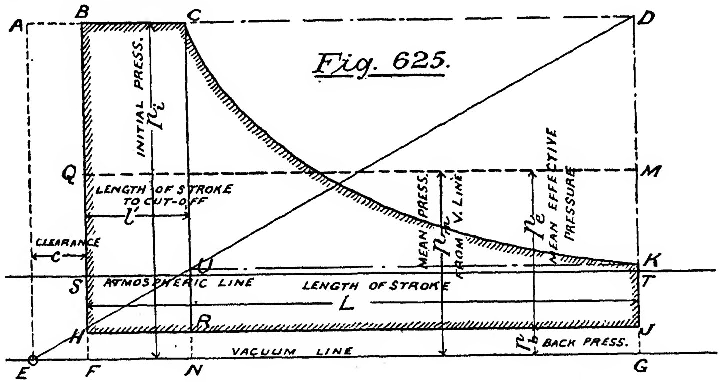
Strokes being equal, the area, or diameter squared, will *represent* cylinder volume. The squares of the diameters are as 4 . 10·89 : 29·2. Taking clearance at $\frac{1}{8}$ cylinder volume for the H. P., $\frac{1}{10}$ for the I. P., and $\frac{1}{11}$ for the L. P., they are represented by ·5, 1·1, and 2·65. In the large diagram, set up at MA a scale of absolute pressures per sq. in., and measure volumes along OK to *any* convenient scale. Thus the dotted rectangles CE, ZH, and QU are obtained, in which the indicator diagrams are to be inserted. Divide DE, GH, and JK, each into 10 parts, and erect vertical lines, upon which pressures are to be placed, as taken from corresponding lines on the diagrams W, X, and Y, being careful to set them up to *absolute* scale; and the shaded curves are obtained.

Next mark point of cut-off B, from which to draw the saturation curve. The latter being always shewn in terms of specific volume (see Fig. 608), divide AB into 2·7 parts, or the volume of one pound weight of steam at 165 lbs. absolute pressure. The method of division is shewn at ML: an inclined line is drawn and 2·7 divisions to any scale placed upon it; then parallel lines to ML will divide the latter proportionately. The volume ·41 cub. ft. has thus been found, which being crossed by ·35 lbs. sq. in. minus, gives the new origin for the curve BSR, to be drawn as a hyperbola in the usual graphic manner (Fig. 620). A second curve CTU may be traced by dividing AC into 2·7 parts and proceeding as before, the origin being then much nearer O.

By stepping the cut-off ML into the whole volume MK, the number of total expansions 13·84 is found, the pound weight of steam occupying at the end of the low-pressure stroke a volume of 13·84 × 2·7 or 37·4 cub. ft. The shaded areas, then, further represent the work done by one lb. weight of steam, if the base lines be specific volumes, and the pressures taken from the

pressure scale, but multiplied by 144 to obtain pounds per sq. ft. The gaps between areas and saturation curves show work lost, but while there is a loss on the side S, there is a gain on side T. A much greater loss occurs from initial condensation, which is not here shewn, and the total 'missing quantity' can only be discovered by some construction such as is given at p. 764. Also it must be understood that the saturation curves cannot be exactly followed except where good steam jackets are adopted; the curve should otherwise be nearer Rankine's adiabatic $PV^{\frac{10}{9}} = C$, which falls slightly below the saturation curve.

Calculation of Work and Horse Power from Theoretical Indicator Diagram.—It is sometimes convenient to make rough preliminary calculations from a simple



Theoretical Indicator Diagram.

hyperbolic diagram as in Fig. 625, where various losses at the corners, caused by release, wire-drawing, cut off, &c., are neglected. Then

$$\text{Area of BN} = p_i\, l'$$

$$\text{Area of CKNG} = p_i \,V \log_e r = p_i\,(l' + c)\left(\log_e \frac{L+c}{l'+c}\right)$$

$$\text{Mean effective pressure } p_e = \{(\text{areas BN} + \text{CKNG}) \div L\} - p_b$$

$$= \frac{p_i}{L}\left\{l' + (l'+c)\left(\log_e \frac{L+c}{l'+c}\right)\right\} - p_b$$

Then, $$\text{Horse Power} = \frac{p_e L a N}{33{,}000}$$ (*See Appendices I. and III., pp.* 772 *and* 931.)

Note that N = *strokes* per min., and logarithms are hyperbolic.

HYPERBOLIC OR NAPIERIAN LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
1	0	3·5	1·252	6	1·791	8·5	2·140
1·25	·223	3·75	1·321	6·25	1·832	8·75	2·169
1·5	·405	4	1·386	6·5	1·871	9	2·197
1·75	·559	4·25	1·446	6·75	1·909	9·25	2·224
2	·693	4·5	1·504	7	1·945	9·5	2·251
2·25	·810	4·75	1·558	7·25	1·981	9·75	2·277
2·5	·916	5	1·609	7·5	2·014	10	2·302
2·75	1·011	5·25	1·658	7·75	2·047	12	2·484
3	1·098	5·5	1·704	8	2·079	15	2·708
3·25	1·178	5·75	1·749	8·25	2·110	18	2·890

Diameter of Cylinder for given Ind. Horse Power may be deduced from the formula already given for the latter. Thus:

$$\frac{p\,\mathrm{L}\,\frac{\pi d^2}{4}\,\mathrm{N}}{33{,}000} = \mathrm{I.\,H.\,P.}$$

$$\therefore d = \sqrt{\frac{\mathrm{H.\,P.} \times 33{,}000}{p\,\frac{\pi}{4}\,\mathrm{L.\,N}}} = 205\sqrt{\frac{\mathrm{H.\,P.}}{p\,\mathrm{L.\,N}}}$$

Horse Power in terms of Steam used. Piston area being measured in square feet, $(l' + c)\,\mathrm{A}$ = volume of steam up to cut-off, and $(l' + c)\,\mathrm{AN}$ = cubic ft. of steam used per minute. But if area be in square feet, steam pressure must be measured per square foot in the horse power formula; also

$$\frac{\mathrm{L} + c}{l' + c} = r \qquad \text{and } \mathrm{L} = r\,(l' + c) - c$$

$$\therefore \mathrm{I.\,H.\,P.} = \frac{\mathrm{PLAN}}{33{,}000} = \frac{144\,p\,\{r\,(l' + c) - c\}\,\mathrm{AN}}{33{,}000}$$

$$= \frac{144\,p\,\{r\,(l' + c)\,\mathrm{AN} - c\,\mathrm{AN}\}}{33{,}000}$$

$$= \frac{144\,p\,(r \times \text{steam c.f. per min.} - c\,\mathrm{AN})}{33{,}000}$$

Such a method of reckoning horse power is convenient when deciding boiler capacity and heating surface. Then:

$$\frac{\text{Steam per minute}}{\text{in cubic ft.}} \Big\} = \left\{\frac{(\text{I. H. P.})\ 33{,}000 + 144\,pc\,\text{AN}}{144\,pr}\right\} \text{D}_{\text{f}}\,\text{L}_{\text{q}}$$

$$\frac{\text{Steam per minute}}{\text{in lbs.}} \Big\} = \left\{229{\cdot}16\,\frac{\text{I. H. P.}}{pr\text{V}} + \frac{c\,\text{AN}}{r\text{V}}\right\} \text{D}_{\text{f}}\,\text{L}_{\text{q}}$$

where V is specific volume at the higher temperature, D_f the diagram factor (p. 772), and L_q the liquefaction factor (*See Appendix II., p.* 884). If steam per brake horse power be desired the value B. H. P. $\div \eta$ may be inserted instead of I. H. P., where η is the mechanical efficiency. Willans' important law connecting steam consumption and H. P. is given in Appendix II., p. 892.

In the above formulæ p is mean effective pressure per square in. At p. 625, this quantity is estimated in terms of initial pressure. If then it be required to know the volume of steam used, in terms of the initial pressure, it is only necessary to substitute the value at p. 625 for p. (*See Appendix III., p.* 931.)

General idea of the various forms of Steam Engine.—The steam engine is a prime mover designed for converting heat into work by allowing steam to expand behind a working piston. Sometimes the work need only be of a reciprocating nature; while in other cases, and this by far the greater number, rotative motion is required, and the crank and connecting rod, or some similar appliance is then employed, as fully set out at pp. 486 to 496. Sometimes also a rotative shaft is introduced, with a fly-wheel to assist in maintaining regular reciprocating motion, where that only is needed, or perhaps to work the valves.

The *Beam Engine*, though almost obsolete, has served and is serving much useful purpose, and a few of its applications will therefore be described. In Fig. 626, A is a Cornish pumping-engine, *a* being the cylinder, *e* the working beam, and *f* the pump-rod passing down the pit-shaft. Steam is that known as 'low pressure,' having only a few pounds' pressure above the atmosphere; and there are three drop valves, *b*, *c*, *d*, for its distribution, called respectively the steam, equilibrium, and exhaust valves. The last passes the steam into the condenser *g*,

where a vacuum is formed and maintained by the action of the air pump *h*. Fig. 608 shews that water under low pressure (as in a condenser) will boil and form vapour at a low temperature; and the air pump has to remove this vapour as far as possible, as well as the condensation water. Even then there is always a back pressure of 3 or 4 lbs. per sq. in. When the piston descends, valves *b* and *d* are open and *c* closed, there then being boiler steam at top and a vacuum below; during the upstroke, *b* and *d* are closed and *c* is open, which places the piston in equilibrium, when the pump rods raise it by their weight. The parallel motion (Watt's) is explained at p. 499; but in A, Fig. 626, one radius link is formed by the portion *e k* of the beam, and a parallelogram then connected to the middle link *k b*, so that the valve and piston rods move on parallel lines.

A rotative beam engine is shewn at B. It differs from A in having the crank and connecting rod instead of pump rod, and four drop valves instead of three, the reason being that each end of the cylinder must now be connectable with boiler or condenser at will, and must therefore have a steam and exhaust valve. The method of distribution is given in Fig. 629, where the left pipe admits live steam to either end of cylinder, and the right pipe similarly removes the exhaust steam, whenever the proper valves are lifted.

A direct-acting pumping engine like that at C may have a beam solely for actuating the valves and air pump, though it also serves to guide the piston-rod. The straight line motion is Scott-Russell's (see p. 486), sometimes called 'grasshopper' gear. A beam blowing engine is shewn at D, *a* being the steam cylinder and *b* the blowing cylinder, the latter having inlet valves *d d*, and outlet valves *e e*, for both ends, so that the issuing air may pass continuously to the blast furnace or other place of use. The fly-wheel is introduced to steady the motion.

E is a compound beam engine. The high-pressure cylinder *a* is placed nearest the beam trunnion, and the low-pressure cylinder further outward. The valves are not shewn, but are so arranged that, when the steam has done its work in the H.P. cylinder, it is allowed to expand into the L.P. cylinder before passing to the condenser.

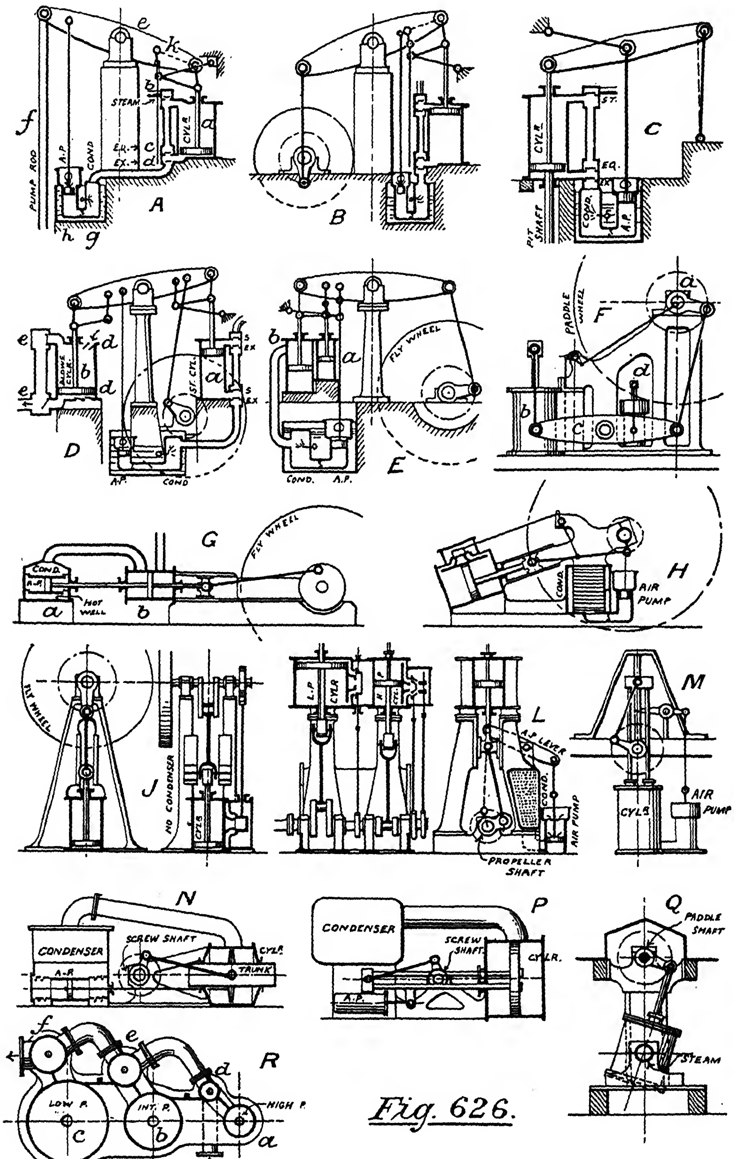


Fig. 626.

Outlines of various Engines. (SLOW AND MEDIUM SPEED)

The side lever marine engine F, the first form considerably adopted on steamboats, was but a beam engine doubled upon itself so as to save room. *a* is the paddle-shaft, *b* the steam cylinder, *c* the beam or 'side lever,' and *d* the air pump.

The *Direct-acting Engine* is shewn in various forms in diagrams G to R, Fig. 626. G is a horizontal factory engine, with condenser *a* behind a cylinder *b*, so that the air pump may be worked in a simple manner by projecting the piston-rod backward. By dispensing with the beam very considerable friction at the trunnion bearing is avoided, caused as such friction was by both load and resistance, or double the piston load. In the horizontal engine there is, however, some additional frictional loss, due to weight of parts and thrust of connecting-rod, while in the vertical engine, although the former is eliminated, the latter still remains.

The diagonal paddle engine at H, like other marine engines, is designed to save room. Whenever paddle propulsion is employed, these engines are now chosen for the purpose. The condenser and air pump are placed within the 'triangle.' J is a form of factory engine seldom employed, but given as an example of a vertical engine with cylinder at bottom and crank overhead; the slide valve replaces the four drop valves of Fig. 629, being worked by eccentric from the crank shaft.

Two other paddle engines are shewn at Q and M. Q is the oscillating engine, exceedingly simple so far as the main mechanism is concerned, dispensing with a connecting rod; but the valve gear is more complicated than with fixed cylinders. The steeple engine (M) was introduced to save head room in shallow boats. Two piston rods are employed, and the paddle shaft is placed between crosshead and cylinder; the connecting rod is said to be 'returned.' The principal objections to this design are the difficulty of staying the slide bars, and of keeping two parallel glands steam tight.

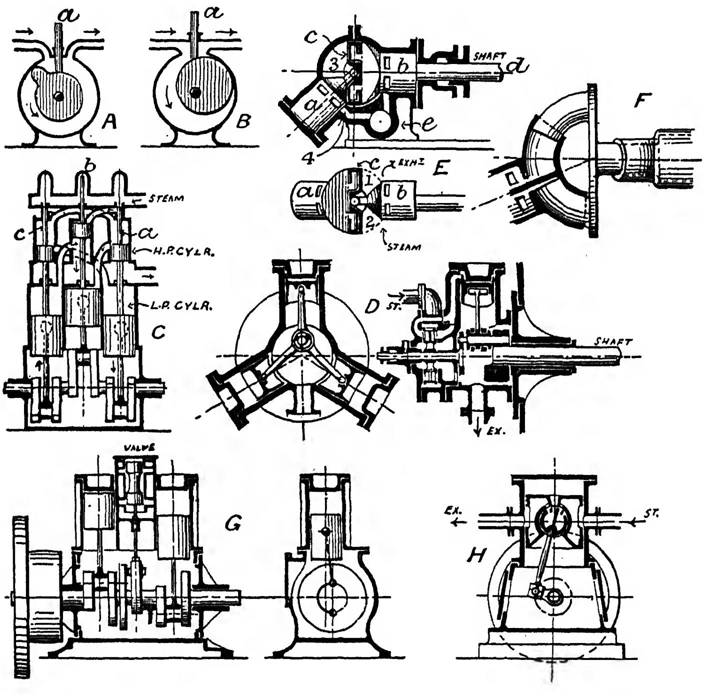
The Penn trunk engine (N) and Maudslay return-connecting rod engine (P) are examples of early screw engines. Being both placed athwart the ship, they must be shortened in length as much as possible. Penn got rid of piston rod length by using a trunk piston and driving the air pump by a rod connected directly

to the latter. The practical objections were the difficulty of packing the necessarily large glands, and of getting at the trunk pin; but a more serious objection was the increased cooling surface. Maudslay's engine was essentially the steeple engine laid horizontally, the air pump being worked from a projection on one of the piston rods. The packing of the parallel glands was the only difficulty.

The modern marine engine is always either compound, triple, or quadruple in design, the two-cylinder compound being shewn at L, which also serves to explain the triple or quadruple. The type is known as the 'vertical inverted,' or 'steam hammer,' and is merely a direct-acting vertical engine with cylinder above and crank below, to give sufficient propeller immersion with direct driving. The slide valves are driven by eccentrics as at J, and the air pump by a rocking lever. The surface condenser is cast with the standards, on one side, and the exhaust steam sometimes passes through one of the standards; but the method is not advised by some engineers, because of irregular alignment caused by expansion. When the triple engine is adopted, the valves are either placed between the cylinders, or as at R, on one side. In the latter case the valve gear must be somewhat altered. *a*, *b*, and *c* are the cylinders seen in plan, and *d*, *e*, *f* the respective valves: in this example of piston form. The passage of the steam will be understood from the sketch, entering first through *d* to *a*, then through *e* to *b*, through *f* to *c*, and finally out to the condenser.

High-speed Engines are a class of engine, usually of small proportions, making 500 revolutions per minute or more. A few principal examples are given at Fig. 627. A and B are types of the *rotary* engine, much in favour with inventors about the year 1870 and previously, but now practically discarded. A may be called the 'annular' and B the 'eccentric' type, a sliding 'abutment' *a* being required in each case to receive the reactionary pressure. There were difficulties in these engines regarding packing and expansive working. Willans' side-by-side three-cylinder engine C, and Brotherhood's three-cylinder engine D, dispense with valve gear. At C the piston rods *a*, *b*, *c*, act as valves, each admitting or cutting off steam to the next high-

pressure cylinder in order. The high-pressure pistons further act as valves for similarly distributing steam to the low-pressure cylinders. Engine D has a special valve of annular form, through which the steam passes both to and from the cylinders, as shewn by arrows. C and D are single-acting engines, so far as each



Sections of various Engines (HIGH SPEED) *Fig. 627.*

cylinder is concerned, the steam pressure being felt only on one side of the piston; but, taking the three cylinders together, there is an impulse every third of a revolution, instead of every half revolution, as in ordinary single-cylinder double-acting engines. In both engines it is only necessary to turn on steam to start in any position, while if reversal is required, an extra four-way plug-cock, called a reversing valve, is interposed, whose duty is to

change the order of the passages, making the steam the exhaust passages and *vice versâ*.

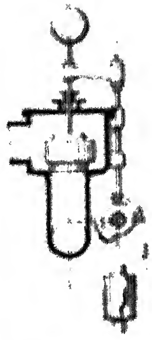
The Tower spherical engine, E, and the Fielding engine, F, are kinematically based on Hooke's joint (Fig. 475, A). In the former two revolving bodies, *a* and *b*, are hinged on opposite sides of a central disc or 'wobbling' piston C, the hinges being at right angles to each other. Within the hollow sphere are four divisions, 1, 2, 3, and 4, the last shewn closed. As the bodies *a* and *b* rotate, and the disc *c* wobbles, the divisions will in turn open and close; and it follows, conversely, that when steam is admitted to these chambers consecutively, the said movements of the disc and bodies will be imitated, and the shaft *d* rotated. To effect this, steam is admitted on one side of the supporting web *e*, passed through proper ports to the four divisions in correct order, and exhausted on the opposite side of *e*. The Fielding engine works similarly, the practical difference being that four curved cylinders are employed, instead of quadri-spherical chambers, corresponding pistons being formed on the central disc. A larger obtuse angle between the inclined axes probably reduces the frictional loss.

The Westinghouse engine, G, is a type of many modern high-speed engines, two single-acting pistons forming the equivalent of one double-acting engine. A piston valve distributes the steam, and the alignment of piston and crank should be noticed. The down-stroke only being of importance, the cylinder centre-line splits the crank *radius* instead of the crank *circle;* the connecting-rod's angular vibration on down-stroke is therefore halved, and a much shorter rod may be employed, securing compactness. During the up-stroke the rod is at a bad angle, but that is of no consequence. The Newall engine, shewn in section at H, is exceedingly interesting, through dispensing with so many working parts; in fact, greater simplicity with efficiency could scarcely be conceived. There are two sets of rings on the trunk piston, between which are slotted holes for the passage of steam. The distribution is effected by enlarging the trunk pin or connecting-rod end into a hollow valve, with a partition; and ports are so arranged that steam is admitted to, or exhausted from, the back of the trunk, at correct times, merely by the vibration of the connecting rod. (*See pp.* 893, 966, *and* 1138.)

Distribution of Steam in Cornish Engines. As no rotative shaft is employed, the valves must be lifted by means of some exterior device, the apparatus usually adopted being known as the cataract (Fig. 628). G is the steam, H the equilibrium, and J the exhaust valve. L is the cataract for opening the steam and exhaust valves, and M that for the equilibrium valve, while A and B are the respective 'plug' rods, worked from the beam. Supposing A to move downwards, the roller R catches the cataract lever D, and raises the pump plunger E, drawing in a large volume of water through the suction valve F. Meantime the valve lever P is held by the stop-piece K on the plug rod, and further secured by the catch lever S, holding the quadrant, so that valve G remains closed.

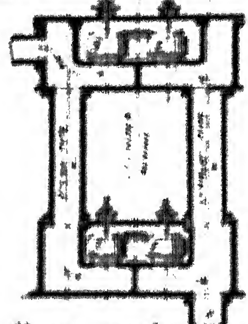
The plug rod now returns upward, and the weight C acting on lever D endeavours to push the water out of the pump into the tank; as it cannot pass by the suction valve, it must leave by the cock V, which admits of regulation, and thus the speed of fall of D, or rise of E, may be accurately adjusted. The lifting rod B travelling upward, will strike and raise the catch levers S and T at any appointed time, and the plug rod then being at the top of the stroke, the valve lever P is free to rise by a left-handed turn, as soon as released, the actual movement being caused by the falling weight Q, and thus the steam valve G and exhaust valve J are lifted by the cataract L. In like manner cataract M governs the opening of the equilibrium valve I, which, it will be remembered, is to be open during an opposite phase of stroke.

Double-acting Engines with Drop Valves. It has been already mentioned that two steam and two exhaust valves are required for these engines. Fig. 629 is a vertical and Fig. 630 a horizontal arrangement, the pipes being connected to a 'nozzle box' at each end of the cylinder, in each of which a steam and exhaust valve may be lifted at the required time by automatic valve gear. An eccentric usually actuates the exhaust valve, but the steam valve is worked by cam or some form of trip gear. The former arrangement is shown in Fig. 631, the shape of cam being such as to open the valve through a small portion and close it during a large portion of the stroke. Sliding the cam on its shaft (in plan) will vary the cut off.



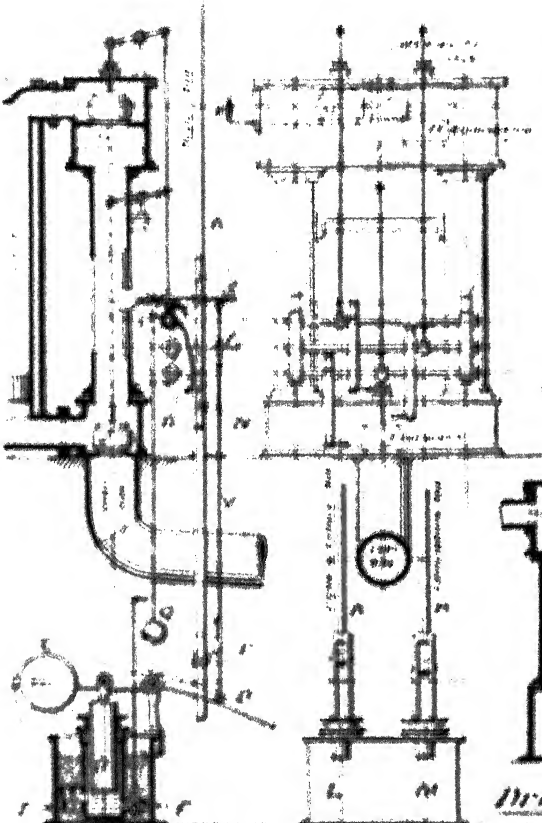
Drop Valve

Fig 631

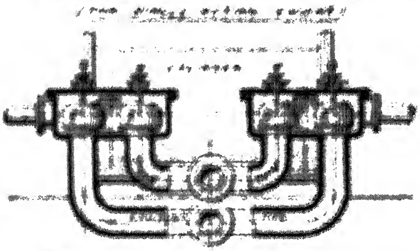


Drop valves

Fig 629



Cataract Valve Gear Fig 628



Drop valves Fig 630

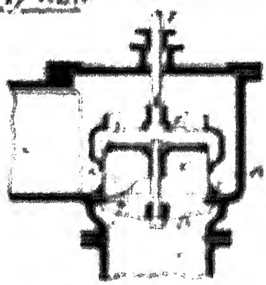


Fig 632

Cornish Valve

The form of Grey's valve, known as the [illegible] valve, is given in Fig. [illegible], and is there shown [illegible] to clearly distinguish the valve from the seats. The steam is [illegible] as entering from below, and while the [illegible] seat [illegible] the upper one consists of a plate [illegible] and bolted to the bridge [illegible]. As [illegible] covers the opening, though at a higher level, the valve, when [illegible], is entirely shielded from the steam pressure below, so far as that pressure tends to lift or depress the valve, and the latter is therefore only the recipient of [illegible] total pressure. Consequently the valve is wholly 'balanced,' that is, the rod [illegible] has merely to lift the dead weight. On account of distortions caused by unequal expansion, the valve should be finally ground on its seat while hot.

Distribution of Steam by Slide Valve. Murdoch (Watt's manager) substituted a single slide valve for the four [illegible] valves in the double-acting engine, the early form being [illegible] 'long D,' so called because it took the shape of a [illegible] in section, the flat towards the cylinder. This was [illegible] the 'short D' valve so well known nowadays, and [illegible] in Fig. 6[illegible]. Its position in the cylinder is shown in Fig. [illegible], where A is the piston sliding in the cylinder [illegible], [illegible] the piston rod, [illegible] the crosshead, and [illegible] the connecting rod, [illegible] the crank, [illegible] the valve spindle, and [illegible] the slide valve, [illegible] the steam chest, and [illegible] the steam pipe, [illegible] and [illegible] the steam ports, and [illegible] the exhaust port. The valve is just opening to steam by the port [illegible], to move the piston to the right, while the exhaust passes by port [illegible], through the valve chamber and [illegible], to the exhaust pipe. When [illegible] has completed its stroke, [illegible] will have been automatically moved to the left, thus admitting steam on the right side of the piston, and exhausting on the left side, causing the return stroke. The [illegible] in the valve allows the latter to adjust itself to port face after wear.

Lap of Slide Valve. The 'normal' valve, Fig. 6[illegible], being that shown hatched only, *just* covers the steam ports when in mid-stroke. Such a valve admits full steam during a whole forward stroke, and exhausts during the whole of the return stroke. An early cut-off is obtained by the addition of lap, the black patches U and V being known as outside or STEAM LAP, and those at W

and X as INSIDE OR EXHAUST LAP, *forming an additional width to the valve face, in line with valve spindle, on the steam or exhaust edges of the valve respectively*, for the purpose of giving early cut-off to steam or exhaust. By adding steam lap the width of opening is decreased, which is, however, compensated by giving increased travel to the valve. Inside lap is rarely necessary, the alterations in valve position caused by introducing steam lap usually giving a sufficiently early cut-off to exhaust (compression point). Various interesting points are raised by altering the proportions of the slide valve, which will be fully investigated later. (*See p.* 772.)

Relation of Crank and Eccentric.—The commonest valve gear is the eccentric and rod. The eccentric is merely a convenient form of crank whose pin is so enlarged as to envelop the shaft: it follows that the eccentricity or length of *eccentric crank* must be measured from centre of eccentric sheave to centre of shaft. This amount we shall sometimes call the *throw*. While, then, the piston moves the crank, the latter in turn moves the eccentric, and so automatically, by the slide valve, adjusts the supply of steam.

(*Without lap.*) A normal valve must of necessity be at *half stroke* when the piston is at the end of its stroke—that is, when the crank is at a dead centre; for then the valve should be just opening to steam. The eccentric crank must therefore be placed at 90° to the engine crank. Further, the direction of rotation will be determined by the position, right or left of it, of the eccentric. The eccentric will always *lead the crank* or travel before it; for, if we endeavour to turn oppositely, we shall only close the steam port at the very time it should be opening, and so block the supply. Therefore, *in a normal valve, the eccentric must lead the crank by* 90°.

(*With lap.*) Let us next consider a valve having lap. Referring, again, to Fig. 634, the thin outline shews a valve with lap, placed at mid-stroke. It then covers the steam port plus the lap. The crank being on dead-centre F, it follows that, in order to admit steam by port R, valve must be moved bodily to the right, and the eccentric *lead the crank by* 90° + *lap*, as at H_1. A little consideration will shew that strictly similar conditions obtain with the crank on the dead-centre Z.

(*With lap and lead.*) To assist the compression steam in preventing a knock on the crank at the end of the stroke, it is advisable that the valve be slightly open when the crank reaches the dead-centre. This is called *lead*, and is *the amount of opening of steam port at the commencement of the stroke.* When a valve, then, is provided both with lap and lead, the eccentric must *lead the crank by* 90° + *lap and lead*, the lap only being apparent on the valve, while both are apparent in eccentric position.*

Reversing Gear.—Factory engines always rotate in one direction, and thus only require a fixed eccentric. Again, changing eccentric from H to J, Fig. 634, will change the direction of motion, then shewn by the dotted arrow instead of by the full arrow. Fig. 635 gives a means of moving the eccentric to the opposite position, when the engine is at rest. Sheave B being firmly bolted to a fixed plate A, can, on unloosing C, be slid from *h* to *j* and rebolted, or, still further, can be made to take any intermediate position between *h* and *j*, giving a variation of travel with the same lap. Such decrease of travel means earlier cut-off, as we shall see later.

Reversing by Loose Eccentric.—But it is not always convenient to stop the engine for any considerable period, and Fig. 636 shews one of many methods by which a single eccentric may be quickly changed from one position to the other. C is the crank, having a stop D fixed symmetrically, and A the eccentric sheave, which, being loose on the shaft, is provided with a balance weight E to prevent spontaneous movement. At present the sheave has its centre at *j*, and while the *eccentric leads the crank*, the *crank drives the eccentric;* so, although *j* causes the crank to turn round left-handed, it is at the same time pushed before the crank by the stop D. But the sheave may be swung round to *j* or *h*, when starting the engine, in a manner to be described. Lifting the gab F from the valve spindle pin disconnects eccentric from slide valve K, when the latter may be moved by the hand lever H. On starting, then, the left hand lifts the handle G, while H is moved by the right hand, and thus steam may be admitted at will to either side of the piston, according to the direction in

* The student must carefully distinguish between the two applications of the term 'lead,' which need not, however, create confusion.

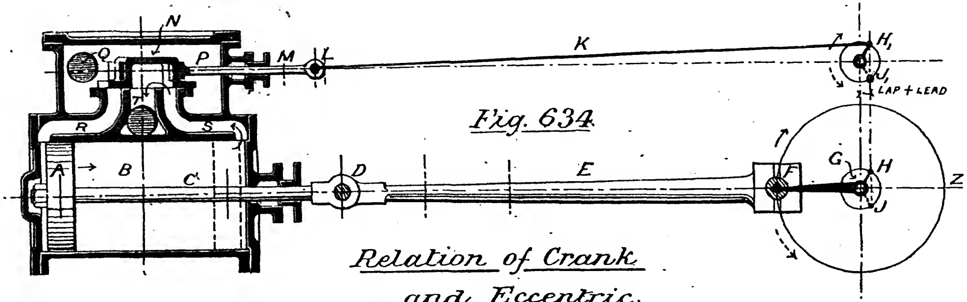
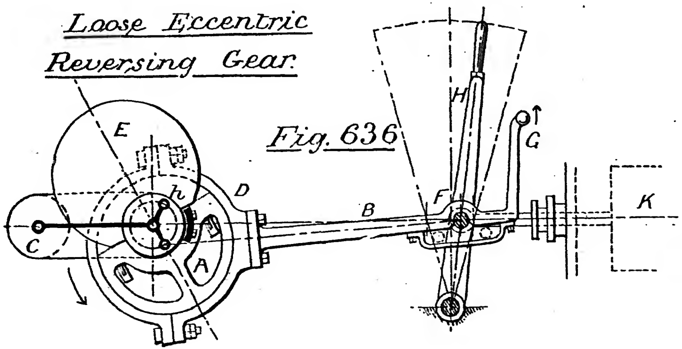


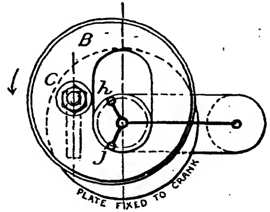
Fig. 634.

Relation of Crank and Eccentric.



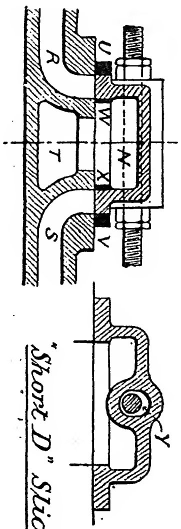
Loose Eccentric Reversing Gear.

Fig. 636.



Shifting Eccentric for variable cut-off or reversal.

Fig. 635.



"Short D" Slide valve.

Fig. 633.

which the engine is to be turned. The slide valve K once opened, G may be dropped, crank C catches up the sheave A by the stop D, F find its way to the valve rod pin, and the gear is once more automatic. The engine may be stopped by lifting the gab.

Reversing by Link Motion.—If two *fixed* eccentrics be placed on the shafts, one for forward and one for backward movement, it can be arranged to put either eccentric in gear as required, the other remaining inactive. The gear for this purpose is known as link motion, and, though more complicated than loose-eccentric gear, is more easily manipulated, and is absolutely certain in action whatever the position of the crank. In *Stephenson's Link Motion*, Fig. 637, the eccentric rods A B are connected to either end of a link C, curved to a radius from D. The valve spindle F supports a die E capable of vertical movement relatively to link C, such movement being controlled by the lifting link G. At present the radius link is in 'mid gear,' and any 'plus' movement of one eccentric rod would be met by a 'minus' movement of the other rod. If these movements were equal, the valve would not travel at all; but, as the sheaves are not placed directly opposite on the shaft, the plus and minus displacements do not balance, and the valve opens to lead.* If the reversing rod H be moved to the right, the rocking link G will lift the radius link until rod B is nearly level with the valve spindle, and the valve then receives almost all the horizontal movement of the B, while A's motion is all but inoperative on the valve. The eccentric B is then in 'full' gear. If H be moved to the left, the A rod is put in gear and B is practically inoperative. J is termed the weigh-bar shaft, and H is coupled to a hand lever on the driver's platform.†

In *Gooch's Link Motion*, Fig. 638, the eccentric rods A and B always vibrate at the same height, and radius link C rocks from a fixed point G. But the valve rod is in two parts, one of which, K, the intermediate valve rod, being lifted or lowered, changes also the position of the die E. In the figure, K is shewn in direct connection with the rod A, while B's vibration has no effect on the valve. When K is at its lowest position, rod B is in gear and A is noperative; link C has it curves struck from D. It should also

* Larger lead than that in full gear.

† In large marine engines it is usual to reverse by steam power.

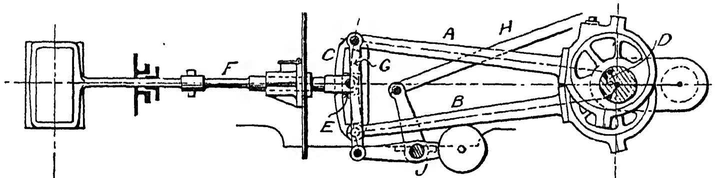
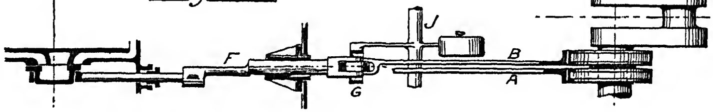


Fig. 637.



Stephenson's Link Motion.

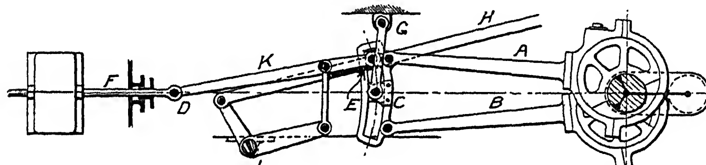
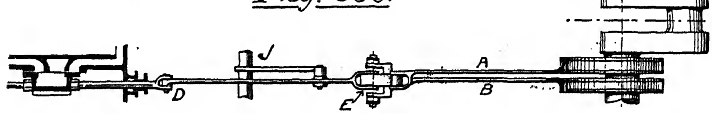


Fig. 638.



Gooch's Link Motion.

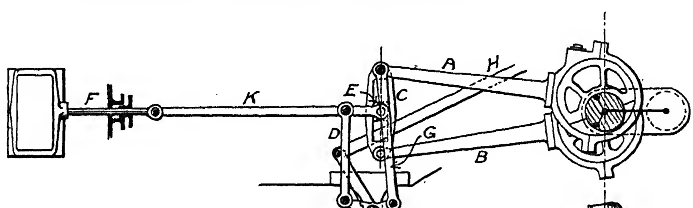
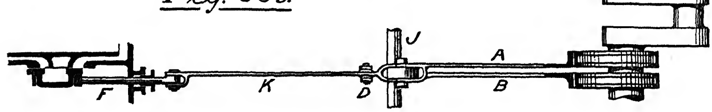


Fig. 639.



Allan's Link Motion.

be noticed that when the [illegible] link [illegible] the [illegible] eccentric rod centre lines [illegible] valve travel and character of [illegible] times, as in Figs. [illegible] and [illegible], the [illegible] larger than the valve travel. [illegible] the link is lifted, as in Stephenson's gear [illegible] the crank, while if the valve rod [illegible] the valve. [illegible], were the [illegible] with a lowering of the valve rod, [illegible] it would [illegible] neither to one side or the other, that is, would be straight. This is obtained in *Allan's Link Motion*, Fig. [illegible], the movement being analogous to Watt's parallel motion. A double armed lever is fixed to the weigh-shaft [illegible], which, being turned by rod [illegible], moves the rocking links [illegible] and [illegible] in opposite directions, so as to bring [illegible] or [illegible] opposite [illegible], as required. [illegible]

Reversing by Radial Valve Gear.—Many [illegible] have endeavoured to obtain a simpler reversing gear [illegible] described, by taking motion from the connecting rod. [illegible] successful is *[illegible] Valve Gear*, Fig. [illegible]. A point [illegible] the connecting rod describes a horizontal ellipse [illegible] being attached to the [illegible] within [illegible] to describe the [illegible] connected to [illegible] at [illegible], and to the [illegible], being [illegible] at its lower end in the [illegible], moves [illegible] and down the curved slot [illegible], [illegible] the [illegible] vertical ellipse [illegible]. The proportions of the links are such that the width [illegible] of the ellipse exactly equals twice (lap + lead). Now, the curved slot [illegible], whose centre of curvature is [illegible], is carried on a weigh-bar shaft having its axis coincident with the point [illegible]. [illegible] there, as [illegible] is vertical, a vertical ellipse [illegible] is formed by point [illegible], and the valve cannot open more than to lead at each end. This is the 'mid-gear' position. Should the weigh-shaft be slightly turned, and the slot therefore inclined, an inclined ellipse [illegible] or [illegible] is formed when it is moved to right or left respectively. The total valve travel would now be represented by the horizontal projections of the ellipses, thus the opening of the steam port would be [illegible], measured horizontally, in addition to the lead. When point [illegible] crosses the weigh-shaft centre, the link takes [illegible]

[illegible]

We have still to show that the moving of [illegible] from right to left [illegible] will reverse the crank motion. Let the crank be [illegible] as shown by arrow; [illegible] will then be [illegible], and, [illegible] being [illegible] in a [illegible] from the ellipse *kl* will be described [illegible] moving from *k* to *l*, and opening the valve to steam at the side of piston required. If, on the contrary, ellipse *mn* had been adopted, the [illegible] would have closed the valve, or the crank could not have moved in the required direction. It can also be shown that when [illegible] is constrained to move in the path *mn*, the crank will turn opposite to that described.

Hackworth's valve gear, Fig. 6[illegible], is another form of radial motion, inasmuch as the valve travel is derived from similar vertical ellipses to those of Fig. [illegible]. The fixed eccentric [illegible] is directly *opposite* the crank web [illegible], and the other end of the rod carries a die [illegible], sliding in a vertical guide [illegible]. A point [illegible] will then describe ellipses such as *mn*, *kl*, Fig. 6[illegible], and the motion of the valve be as previously described. [illegible] is the reversing rod for changing the angularity of [illegible].

A modification of the previous arrangement known as *Marshall's* valve gear is shown in Fig. 6[illegible]. [illegible] is on the opposite side of fulcrum [illegible]; eccentric [illegible] is therefore coincident with the crank web [illegible], and a much smaller sheave results. The pressure on the fulcrum [illegible] is, however, very great, and a vibrating link [illegible], with fulcrum at [illegible], is substituted for the vertical slide. The usual ellipses are described by point [illegible], and the necessary angularity is produced by change of position of the fulcrum [illegible] moving the bracket lever [illegible] to [illegible] by the reversing rod [illegible].

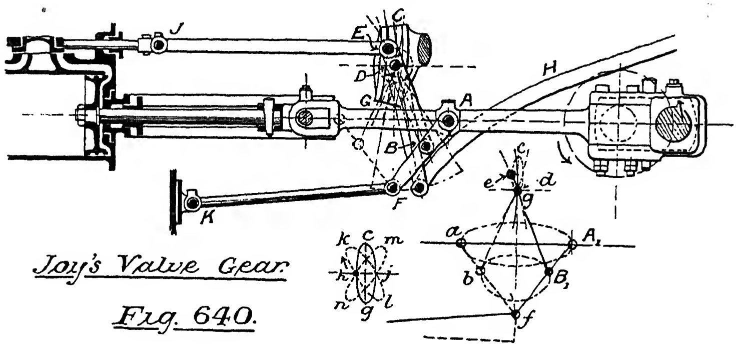
Walschaert's valve gear, Fig. 6[illegible], does not describe the ellipses which are a sign of radial gear, but having some other points in common, is here introduced. The eccentric is fixed at right angles to the crank, as though the valve had no lap or lead; and if connected directly to the valve spindle, would of course only allow the crank to turn in one particular direction.

An intermediate valve rod F, may, however, be changed from D to B, or *vice versâ*, by the reversing lever E, so that F may move either in the same or the reverse direction of J. When F is at B the eccentric must *lead* the crank, as in Fig. 634; but when F is at D, the eccentric must *follow* the crank. The intermediate rod F, again, is only connected to the valve rod G through the lever LM, the pin K forming a fulcrum upon which LM is rocked by the crosshead N. The travel, LP, thus obtained, represents twice (lap + lead), as at *h j*, Fig. 640, and takes effect at the dead centre positions. When F, therefore, is in mid gear at C, the valve opens only to lead, but when moved to D or B, the opening is *eccentric throw minus lap*, as in Fig. 634.

Valve Gear for Oscillating Engines.—The method by which a satisfactory motion of the valve is obtained will now be made clear by reference to Fig. 644. T and U are the valve boxes, of which there are two, in order to keep the cylinder balanced. Y is the cylinder and V W the trunnions, being steam and exhaust pipes respectively, supplied with stuffing boxes. M *m* and N *n* are the valve levers, rocking on fulcra R and S; and P Q the valve spindles, guided at their upper end. All the parts mentioned share in the rocking motion of the cylinder, the remainder are either fixed to the ship or take motion only from the crank. Z Z are fixed guides for sliding link L, whose slot is curved to a radius from trunnion centre. To L is again connected, by centre-pin F, the usual radius link G H, which is moved by eccentrics *c d* through rods D E. X is the trunnion bearing.

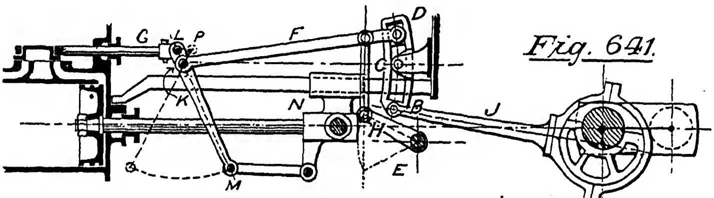
It will be seen that the rocking of the cylinder can in nowise affect the vertical movement of the valve levers; but any motion given by the eccentrics to the link L is faithfully transmitted to the valve spindles through their levers, the discs J K always lying in the link L. On account of the introduction of the rocking levers M and N, the eccentric motion will be reversed. The eccentrics are therefore set to *follow* the crank by 90° *minus lap and lead*, and the rods are said to be crossed.

Steam enters at V, and passes into the valve chests by the belts *e e*, entering through port *a*. After giving work to the pistons through either steam port *b b*, it exhausts through the mid port *c*, and passes out through the belt *f* to the exhaust

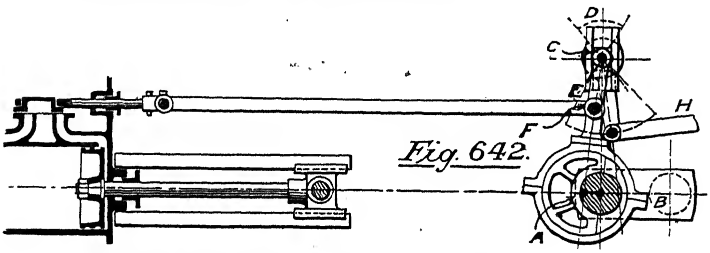


Joy's Valve Gear.

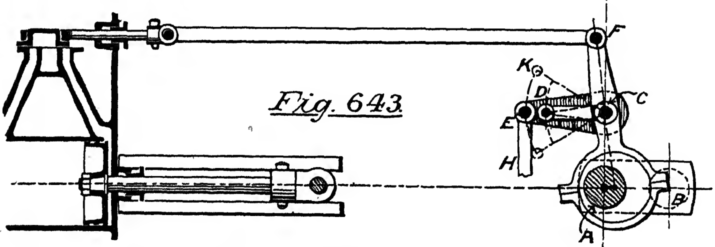
Fig. 640.



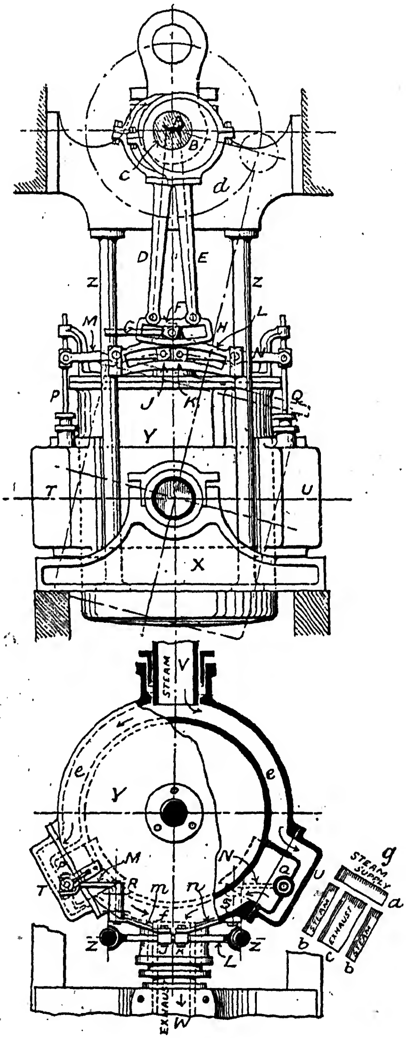
Walschaert's Valve Gear.



Hackworth's Valve Gear.



Marshall's Valve Gear.



Valve Gear for Oscillating Engine

Fig. 644.

pipe W. Sketch *g* is a front view of the ports. A loose eccentric or single fixed eccentric may replace the link motion, but the link L is always required.

The Simple or 'Watt' Governor was invented by Watt for [illegible] the supply of steam to the engines, [illegible] and each [illegible] the balls, which, [illegible] shaft, tended to lift the [illegible] and [illegible] to close the [illegible] valve [illegible] the steam pipe [illegible]. The valve, being of [illegible] the pipe when inclined at about 90° to a cross sectional plane, and is, further, "balanced"; that is, the steam pressure on one half tends to [illegible] it, that on the other half having an opposing tendency.

The function of the governor is to keep the engine speed within reasonably constant limits, whatever the load. The fly wheel obtains approximate uniformity of crank pressure and speed during each revolution, but cannot govern the speed over several revolutions; that is left for the governor, which conversely is unable to control the smaller changes during a revolution. When the speed increases, the balls fly outward and tend to close the valve, throttling the steam supply, which reduces the crank pressure and causes a return to the normal speed. Should the velocity of the balls decrease, the exact converse will happen.

Three forces keep each ball in a radial position, and their proportionate amounts may be determined by force diagram, as at (a). Then $\frac{wv^2}{gR}$, T, w, are respectively proportional to R, l, H.

$$H : R = w : \frac{wv^2}{gR} \qquad \text{and } H = \frac{gR^2}{v^2}$$

$$\text{But } v = 2\pi R n = \frac{2\pi R N}{60}$$

$$h = \frac{gR^2}{v^2} \times 12 = \frac{gR^2 \times 60 \times 60 \times 12}{4\pi^2 R^2 N^2} = \frac{35{,}200}{N^2} \text{ inches}$$

$$\text{And } N = \sqrt{\frac{35{,}200}{h}} = \frac{187.6}{\sqrt{h}} \text{ revolutions per minute}$$

or $h \propto \frac{1}{N^2}$ and $N \propto \frac{1}{\sqrt{h}}$

The Weighted or Porter Governor. [illegible] governor [illegible] on the same [illegible] decrease [illegible] balls [illegible] of its [illegible] force [illegible] increased [illegible] a heavy weight on the central spindle, as in the Porter governor (*b*, Fig. 639), it can be shewn that a given variation of speed of rotation will, under certain circumstances, cause a much greater movement of the sliding sleeve, thus making the governor more sensitive, without, of course, adding to the centrifugal force, speed for speed.

It is customary, in the weighted governor, to make the four arms equal, and the angle *a* approximately equal to the angle [illegible] the rise of W_1 being consequently twice that of W. Now half W_1 pulling at each ball, by the principle of work its effect at [illegible] will be equal to W_1, for $\frac{1}{2} W_1 \times$ dist s = $W_1 \times$ dist [illegible]. We have, then, a total downward pull at [illegible] of $W + W_1$, the centrifugal force remaining as before. Reasoning from force diagram, as at [illegible], we have

$$\frac{H}{R} = \frac{w + w_1}{\frac{w\omega^2}{gR}}$$

$$h = \left(\frac{w + w_1}{w}\right)\frac{gR^2}{\omega^2} \cdot \frac{1}{R} = \left(\frac{w + w_1}{w}\right)\frac{35\,200}{N^2} \text{ inches}$$

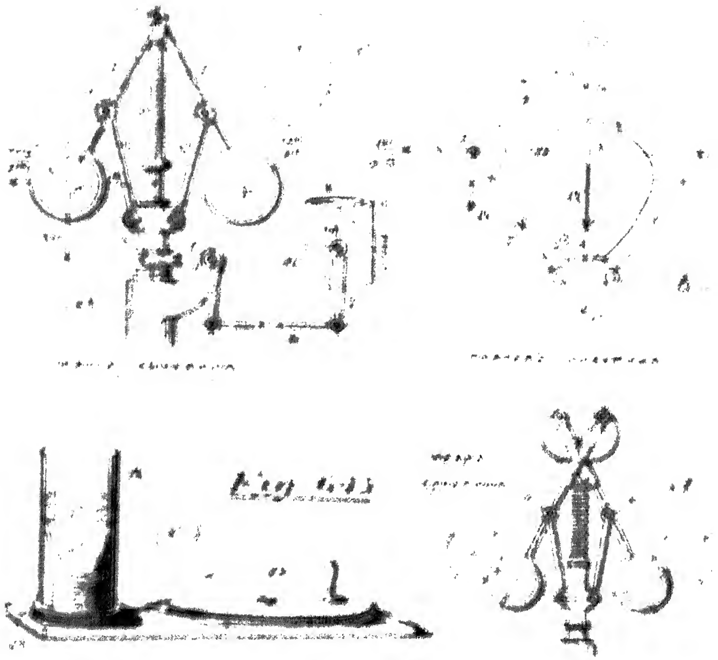
Or, *when the revolutions per minute are the same in both governors, h is greater than in the Watt governor in the proportion of* $\frac{w + w_1}{w}$: 1, and, for considerable change in speed, the variation in height is greater in the same proportion, for

$$h - h_1 = 35\,200\left(\frac{1}{N^2} - \frac{1}{N_1^2}\right) \text{ in Watt governor,}$$

$$h - h_1 = 35\,200\left(\frac{w + w_1}{w}\right)\left(\frac{1}{N^2} - \frac{1}{N_1^2}\right) \text{ in Porter governor.}$$

When, however, the problem is approached practically, its aspect is considerably altered, for it would be impossible to deal

[illegible] suggested. [illegible] heights are equal, [illegible] higher in the proportion of [illegible] heights. The governors will [illegible] with the same increase in speed, [illegible] heights are equal to begin with?



But the *power* of the Porter governor under these conditions will be very much greater, so that the *sensitiveness* shown by theory has simply been changed into the *power* obtained in practice.

If the four arms are not equal, *a* may be supposed to rise one inch, and the rise of *b* be noted. Then the effect of w_1 at *a* will be w_1 × rise of b ÷ [illegible], and H may be found as before. W_1 varies in practice from 60 to 300 pounds, and W from 2 to 4 pounds. Fig. 561, p. 514, is a good example, where the dash pot *r* serves to damp the vertical vibrations.

The "Head" of Water [illegible] [illegible] a [illegible] [illegible] of which is constant [illegible] upon these quantities of water [illegible] balls [illegible] to take care [illegible] correct speed [illegible], so allowing [illegible] [illegible] parabolic path, we must expect them to rise with the least difficulty, and as a practical fact, such a governor, called parabolic, is extremely sensitive, so that the balls will rise or fall to the full extent with a very small alteration in speed. In some governors, the weights are [illegible] parabolic paths, but the usual form is the crossed-arm [illegible] (*d*), Fig. 695, where the ball paths are very nearly parabolic [illegible] spring is placed on the spindle to resist the rise of the [illegible] under small changes of load. (*See* **77**, [illegible])

Variable Expansion Gear. When [illegible] changes, the steam supply must change [illegible] otherwise the engine will either race or stop, and two [illegible] suggest themselves: (a) *throttling*, with constant expansion and altered pressure, by narrowing the steam passage at the supply valve; (b) *variable expansion*, with constant initial pressure, by regulating the movements of the slide valve so as to alter the position of cut-off. Either method may be worked by hand or governor. Judging from the indicator cards alone, and debiting throttling with a loss of energy in the process, variable expansion has always appeared more economical, but as it often involves very greatly increased cylinder condensation, while throttling dries the steam, the supposed advantages are not always great or even present. Captain Sankey's paper before the Inst. Mech. Eng. in 1895, clearly showed the above facts, which he demonstrated by reference to the Willans line. (*See* **177**, **77**, [illegible]) Theoretically, if the pressure increase from [illegible] to [illegible] the gain by expansion varies from [illegible] to [illegible], but

[illegible] respectively [illegible] may come to be deduced from [illegible] that throttling or expansion [illegible] steady [illegible] that vary somewhat slowly, [illegible] expansion [illegible] a decreased consumption in [illegible] engines, and [illegible] engines where load changes rapidly and [illegible]. Besides, that the gain is greater with low than with high pressures, and that expansion gear responds more readily [illegible] changes of load than does the throttle valve governor.

Linking up is a means of obtaining variable cut-off. It may be shown by Zeuner's valve diagram, to be explained later, that a [illegible] travel to a D-slide valve will cause an earlier cut-off to steam, but will also compel an earlier cut-off to exhaust, or compression point, on the back stroke. Referring to Figs. 633 to 639, the mid-gear positions will produce very little motion on the valve, but, when the links are in full gear, the valve will travel its greatest. Any intermediate travel may be procured, and therefore, within limits, any desired cut-off. When, therefore, a locomotive is started, the reversing lever is pulled right over, but when full speed has been obtained, the work becoming less (being that required only to overcome frictional resistances), the driver links up to such a position as to supply just enough steam to do the work. The diagrams obtained are shown at *a*, Fig. 635, 1 being that at starting, 3 with full speed and least resistance, and 2, an intermediate condition. Herein lies one advantage of link motion over the loose eccentric: the former is an economic expansion gear, while variable work must be met in the latter by throttling. The radial gear, Figs. 640, 642, and 643, may also be linked up by turning the curved guides in Figs. 640 and 642, or the lever [illegible] in Fig. 643, through a smaller angle, when the projected width of $k l$ will be smaller, and the valve travel be thereby decreased. One result in this gear has already been referred to: the distance $h j$ is absolutely constant, whatever the position of the curved guides, so the amount of lead never changes, whether in full, half, or mid gear. This is not so with link motion; with 'open' eccentric rods, as in Figs. 633 to 639, the lead is much greater in mid gear than in full gear, and proportionate at other places. With crossed rods (Figs. 644) the lead decreases towards

the central position [illegible] adjusted [illegible]

It is practically [illegible] a large [illegible] engine [illegible] not start in the right direction [illegible] factory engines [illegible] A *link cut-off* [illegible] expansion valve as [illegible] Fig. 64[illegible], for *varying cut-off* [illegible], all the other points on the diagram being governed by the main valve. When the two blocks constituting the expansion valve are rigidly connected, and the variable cut-off is obtained by *altered travel*, the title "link cut-off" is adopted, as in Fig. 648*; but if right and left hand threads are formed on the expansion valve spindle, Fig. 646, and variable cut-off be obtained by turning the latter round, thus altering the [illegible], the arrangement is known as a *Meyer expansion valve*. In Fig. 646, A B is the main valve, having a [illegible] with walls A and B to form ports C and D. The [illegible] can be separated by turning the hand wheel [illegible], which has a square hole in its boss, through which the valve spindle [illegible] passes, attaching the boss to a screw cutting a [illegible], whose movement represents the altered expansion to the ear. When E and F are close together, they are out of gear and cut-off is governed by main valve; but when separated, they cut off steam at the outer edges of the ports C and D. The expansion eccentric leads the crank by 180° in a reversing engine, and rather more in others, and the two valves move in opposite directions at cut-off, thus decreasing wire-drawing.

When the separate expansion valve was first introduced, a separate steam chest was provided, so that after passing through the expansion valve, the steam had to fill the main valve chest before proceeding to the cylinder, thus forming *additional* clearance steam, which would do no work except a small amount during expansion, and, being delivered into the condenser, would tend to increase back pressure. We are not surprised, then, to

* See also Fig. [illegible], p. [illegible].

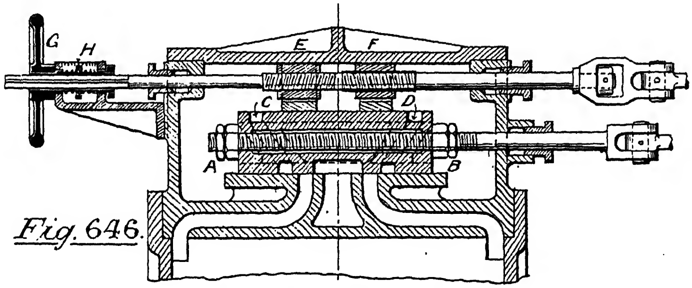


Fig. 646.

Meyer's Variable Expansion Gear.

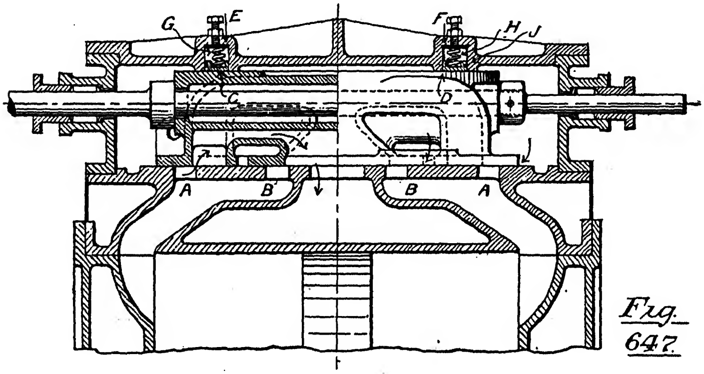
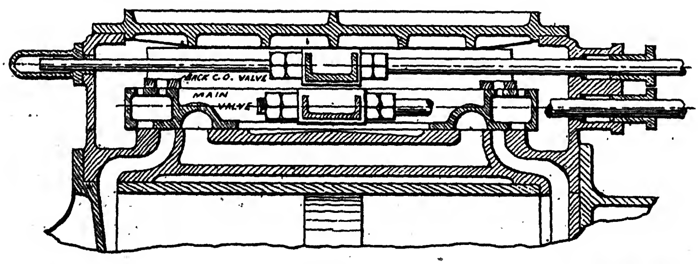


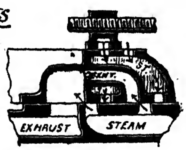
Fig. 647.

Double-ported Slide valve, with relief ring.



Back cut-off valve, with double ports

Fig. 648.



DOUBLE PORTS TO BOTH MEYER & MAIN VALVE

Fig. 649.

learn that very little economy was thus secured. A 'gridiron' valve was adopted for the expansion valve, the ports being split into eight or nine portions, for reasons to be explained in the next paragraph. Clearance is decreased as much as possible in the back-cut-off valve, especially in Fig. 648, though it must always be greater than in a single valve.

A *Double-ported Valve*, as in Fig. 647, is usually adopted for low-pressure cylinders of marine engines. As frictional loss depends directly on distance travelled (total pressure being equal), it is advisable to decrease the travel as much as possible. This may be done by dividing the steam ports into two parts,* as at A B; only half travel is then required. Of course the valve must be made somewhat larger, which increases the total pressure, and consequently the force of friction; so a portion of the back is often shielded or 'relieved' from pressure by the ring C D, which lies in the annular groove E F, being kept steam tight with the back of the valve by the springs G H, and with the groove by the ring J.

At Fig. 648 is shewn a back-cut-off valve with double ports, the main valve being designed to shorten the steam ports and decrease clearance. Fig. 649 shews double ports, both for Meyer and main valve, the arrows indicating the paths of live steam and exhaust.

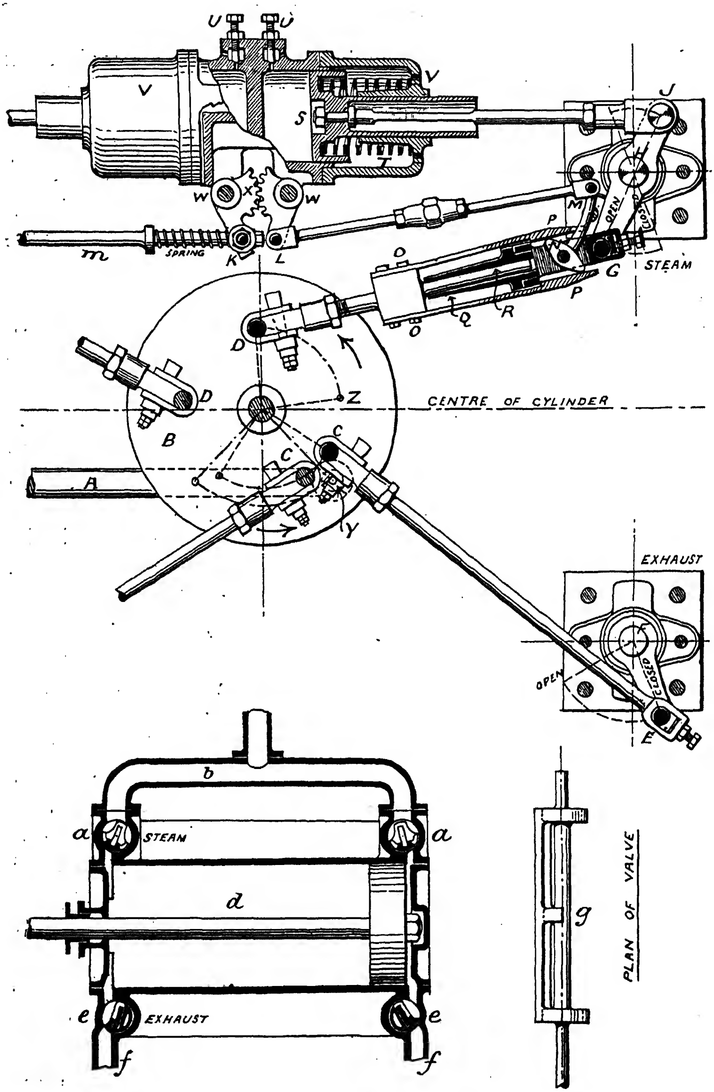
Automatic Expansion-Gear.—Instead of connecting the governor sleeve with the throttle valve, as at (*a*) Fig. 645, it may be allowed to alter the travel of a back-cut-off valve, with increased economy and direct action. The most common arrangement is shewn generally at Figs. 271 and 272, pp. 270 and 271, and in detail at Fig. 261. The expansion eccentric is coupled to the central pin of the radius link, the latter rocking on a pin at the upper end. When the governor sleeve M, Fig. 261, rises, it lifts, by lever H and link K, an intermediate valve rod. Thus the height of the governor decides the height of die in radius link, and therefore the amount of travel on the expansion valve. Eccentric travel remaining constant, when the engine speed increases and the governor sleeve lifts the die nearer the link fulcrum, the travel of the valve is decreased, cut-off is earlier, and less work done. This brings back the speed to the normal.

* Or more, as in gridiron valve.

If, on the contrary, a heavy load is put on the engine, the governor revolving at a low height gives the valve the utmost travel, securing a late cut-off.

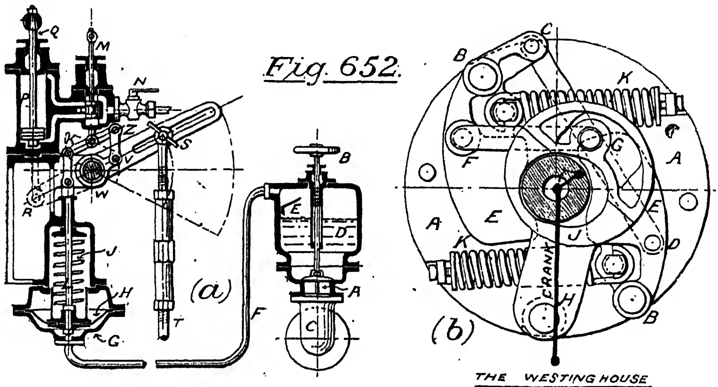
The *Shaft governor* provides automatic cut-off by a very simple and compact arrangement, especially adaptable to small high-speed engines. The gear of the Westinghouse engine is shewn at (*b*) Fig. 652, the object being to directly vary the eccentric throw. A A is a disc fixed to the crank shaft, having pins B B for carrying centrifugal weights E E, and pin H for supporting the eccentric H J. The latter may rock to the right or left on pin H by a limited amount, to be determined by the position of weights E E, their deviation causing an alteration in the eccentric throw. The weights are connected by the link C D, so that their movements shall be simultaneous, and are attached to the eccentric by link F G. If the engines then revolve at a high speed, the weights E E fly outward and pull the eccentric sheave to the left, decreasing throw and producing early cut-off; if the speed decreases, the strong springs K K bring the weights towards the centre and increase the throw. (*See p.* 1145.)

Marine Governors have always been difficult to devise, and, although no perfect governor exists, the arrangement at (*a*) Fig. 652 is one of the best, acting as it does on a direct principle. The fluctuations in speed of a marine engine are caused by the propeller either partly or entirely leaving the water rather suddenly, thus decreasing the load. The consequent increase of speed or 'racing' cannot be ***entirely obviated***, but may be considerably modified by the use of Dunlop's governor. C is a large pipe communicating with the water near the propeller, and D an air chamber which can be shut off from C by the screw-down valve A, worked by hand-wheel B. F is a pipe containing only air, the entrance of water being prevented by the baffle-plate E. H is a diaphragm in communication with F, and L K a rod which partakes of the movement of H, transmitting it to the piston slide valve M, for admitting to or exhausting from cylinder P. The cock L admits steam to M, and the piston rod Q R is connected to the lever R S, which has its fulcrum at W. Finally, S T is a rod for actuating a throttle valve in the high-pressure steam pipe of the engine.



Corliss Valve Gear. Fig. 650.

If the propeller sinks below the normal, water rises in D, and, compressing the air in F, presses on diaphragm H, lifting K L and moving K Z round fulcrum Z. Valve M being opened to steam at the bottom end, piston P is raised, thus depressing the rod S T and opening wider the engine throttle valve. But, as S moves down, the lever K Z is turned round K as a fulcrum, and valve M is once more placed in mid position. Suppose the propeller rises, the air in F becomes more rare, and spring J moves L K downward, opening M at the top, bringing Q R down, and raising S T, thus partly closing the throttle valve. (*See p.* 1144.)



DUNLOP'S MARINE GOVERNOR

THE WESTINGHOUSE SHAFT GOVERNOR

Corliss Valve Gear.—Of all the 'trip' gears,* this is probably best known. In Fig. 650 the upper diagram shews the valve gear, the lower being a section through the cylinder and valve chambers. There are several advantages possessed by this valve arrangement and gear, some being common to other trip gears: (1) a sharp cut-off is obtained, when the 'trip' takes place, preventing wire-drawing; (2) an easier-working form of valve, *g*, is adopted; (3) steam and exhaust parts being separate, there is less loss by initial condensation; (4) clearance is very small; (5) the variable cut-off is automatic.

The valves *a a* admit steam, and *e e* pass the exhaust, being represented in plan at *g*. They are hollow cylinders having a

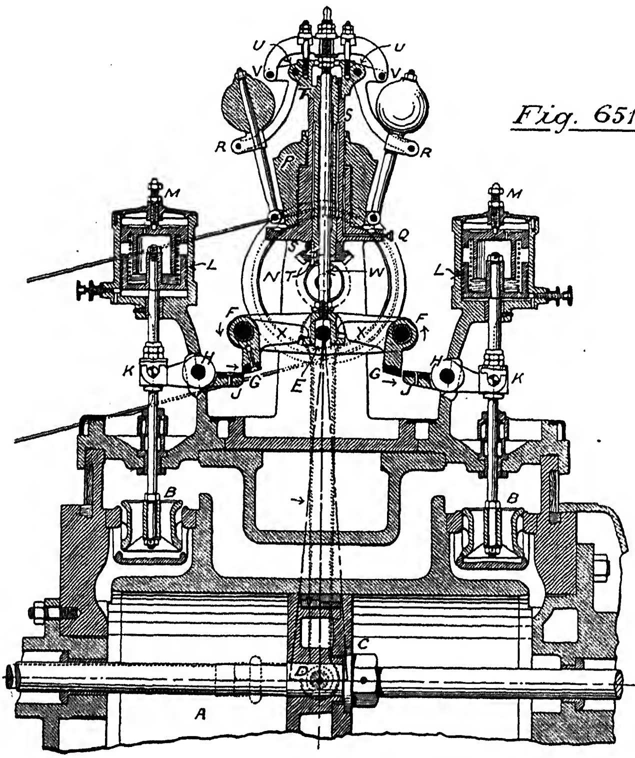
* Term given to rapid cut-off gears, worked by the trip of a valve lever.

large portion cut away, and are rotated by spindles to which they are connected loosely. The steam pipe is shewn at *b*, and *ff* are the exhaust pipes, forming the cylinder supports. Taking the valve gear, A is the eccentric rod, which by a to-and-fro motion rotates the disc or wrist-plate B; to the latter are connected the four valve rods, two of them at C C actuating the exhaust valves, the other two at D D working the steam valves. The exhaust rod C E is directly connected to the valve lever E F, and moves it through rather less than 90°. The steam-valve rod D G is more complicated, consisting of two parts: one, D O R P, attached to the wrist plate; the other, Q N G, connected to the valve lever G J. These tend to separate, by reason of the force in the compressed spring T, but are prevented by the spring catches P P. If, however, the latter are prised apart, spring T is released, and, pulling J rapidly to the left, closes the steam valve. The prising action is obtained by the toe lever M N, which, pinned to Q N G at N, rocks on fulcrum M. As the pin D moves from Z to D, the rod D G takes a more crosswise position relatively to the toe N, and at some intermediate position the catches P P liberate the parts D and G, permitting the valve to be closed. When D moves back to Z, P P regain their normal condition and D and G are connected. The position of fulcrum M determines where, between Z and D, the toe shall release part G, and this is decided by the height of governor sleeve, the latter being connected to rod *m*. When the governors rise, *m* is pulled to the left, moving M an equal amount to the right, levers W K and W L being geared together at X. This causes the toe to separate P P at an earlier part of the stroke Z to D, and the converse will happen when the governors fall. Lastly, the dash-pot S being full of air only capable of passing out at U, reduces the shock caused by the sudden release of the spring T, the set screws serving to regulate the air passage, and the back chamber V is usually connected to the condenser to ensure decision.

The Proell Valve Gear is another good trip gear. The lever D E, rocking on fulcrum E, may for the present be looked upon as rigidly connected to the arm F F, and toes F G. At point E is attached the eccentric rod, and a movement of D to the right will cause the left-hand toe G, trailing along H J, to finally slip, when spring L closes the steam valve B. Meantime, the right-

hand toe, which tripped on the last stroke, must be replaced on J K, and this is attained by making the L-lever X F G free to turn on the pin F, until G is high enough to slip into place The dash-pots are similar to those already described, set screws M M

Fig. 651.



Proell's Valve Gear.

adjusting the compression of the springs L L. A rigid bracket S S supports the governor gear; within it the hollow spindle T T revolves, and the balls, flying outward, pivot R T on T, raising the central weight P, while lifting pins V V and the spindle W to a higher position This rise affects the positions of the toes G G,

bringing them nearer together; the reverse happening when the governors fall, and thus is obtained automatically an early or late cut-off respectively. A dash-pot within P damps small vibrations on the governor, entrance or exit of air being adjusted by screw Q. The steam valves B B, being double-beat, are balanced, besides requiring only half the lift of a single valve.

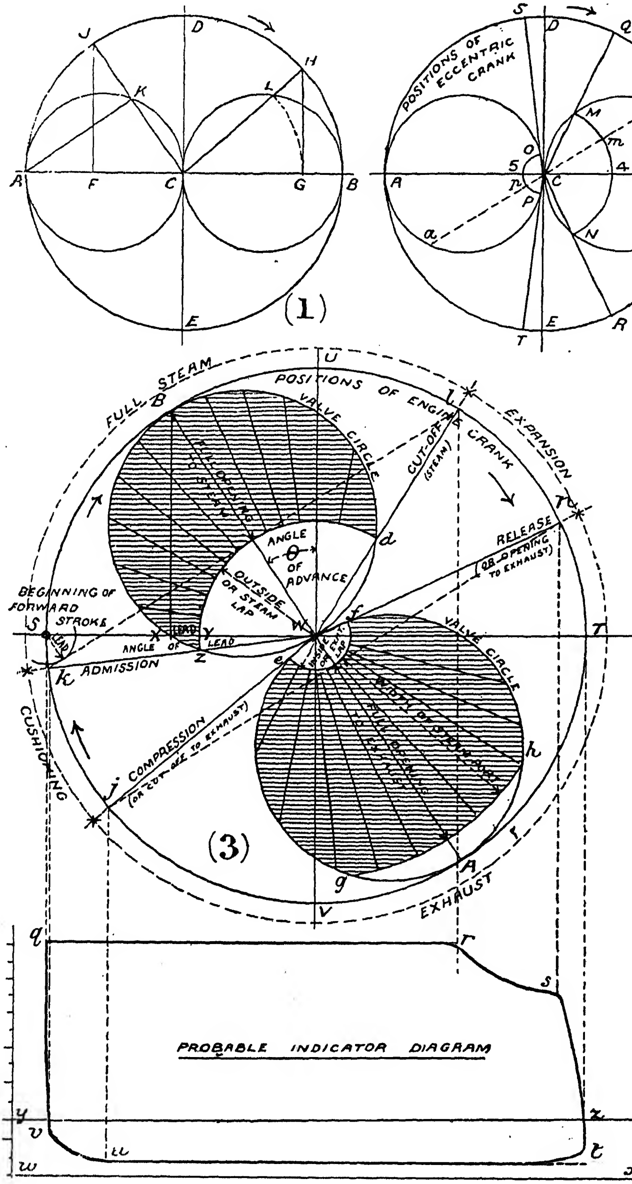
Zeuner's Valve Diagram is a graphic and ready means of finding the various positions of the engine crank where admission, cut-off, release, compression, &c., take place with a D slide valve, when the valve dimensions are known; or, conversely, of finding valve dimensions when certain crank positions are given.

Imagine a valve without lap, and let C D be the eccentric throw or radius at (1) Fig. 653. When the eccentric is at D, the valve is closed, and, when moved to H, the opening to steam *on the left* is C G. Turn G round to L; then C L is steam opening for eccentric position H. A series of points such as L may be found and the curve C L B drawn, whose radii vector shew gradual opening and closing of the left-hand steam port. The left diagram being obtained similarly, join A K. Then triangles C A K, C J F are similar and equal, and A K C is a right angle; the two polar curves are circles, and while circle C B shews *steam* opening, circle C A represents the opening to *exhaust*, together being known as curves of position for a valve without lap.

Taking a valve having lap, both to steam and exhaust, its position curves are those at (2) Fig. 653. For the opening either to steam or exhaust will be that at (1) less the respective lap. At centre C, strike arc M N with radius = lap, while O P = exhaust lap. Then 4 B is full opening to steam at left-hand port, eccentric being at B; Q is admission position, and R that of cut-off. Similarly A S is full opening to exhaust at right-hand port, eccentric being at A; T the release, and S the compression position.

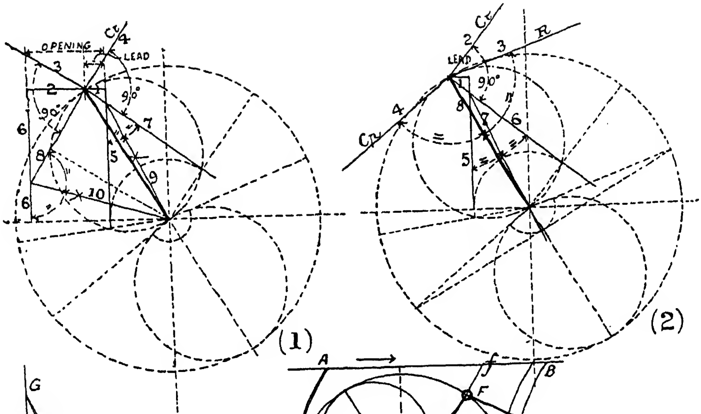
We must now translate eccentric position into crank position. Still assuming a right-handed rotation, we must turn back in a left-handed direction all the eccentric positions, through the angle by which the eccentric leads the crank, to effect the above purpose. This angle is 90° plus the angle of advance.* The change has

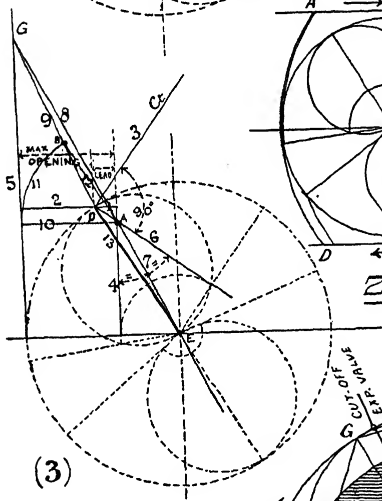
* Angle of advance = the angle whose sine = (lap + lead) ÷ throw.



Zeuner's Valve Diagram, Fig. 653.

SHEWING CRANK POSITION FOR ALL CONDITIONS OF VALVE OPENING.





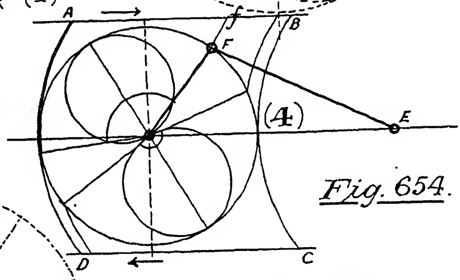
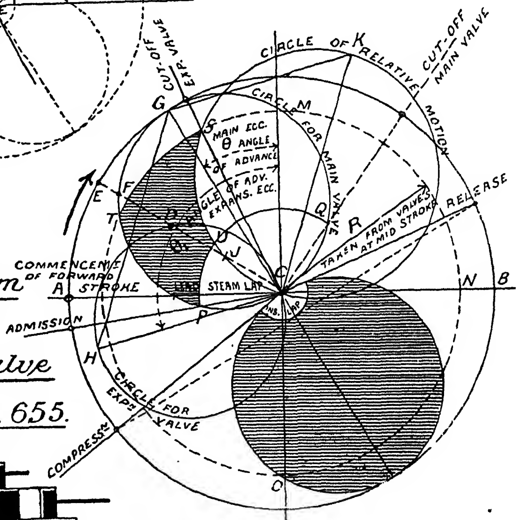


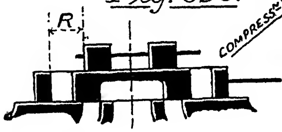
Fig. 654.

Zeuner Problems.



Zeuner Diagram applied to Meyer valve

Fig. 655.



[illegible] diagram, as the width of steam port is not always [illegible] off [illegible]

To find corresponding piston positions, the [illegible] perpendicular, as shown by dotted lines, and dividing stroke [illegible] into [illegible] parts, figure in decimals of forward or backward stroke, as required. But as this is only approximately true for a very long connecting rod, or a slotted rod, the method given in Fig. [illegible] is more accurate. [illegible] being drawn, for forward and backward strokes respectively, and [illegible] the length of connecting rod, and strike arcs from centre point, as [illegible] altered, [illegible] changed [illegible], the [illegible] at [illegible], and the points obtained represent the correct positions of the piston regarding these.

The Zeuner diagram further enables us to find very closely the form of indicator diagram before the engine is built. The method is shown in Fig. [illegible], and may be understood further by [illegible] Fig. [illegible], due regard being also paid to the various losses.

Zeuner Problems. A few problems will now be discussed, where, certain data being given, it is required to draw the whole diagram as in Fig. [illegible]

(1) Given valve travel, both laps, and angle of advance. This is too simple to need explanation.

(2) Given greatest opening for steam or exhaust, both laps, lead, and position of cut-off. Describe travel circle with radius = lap + opening, and arc of lead at [illegible], find [illegible] and join [illegible]. Bisect angle [illegible], and complete.

(3) Given angle of lead, travel, and positions of cut-off and release. [illegible] is found as in last case; then, [illegible] being known, [illegible] may be drawn at right angles to [illegible], and the rest is clear.

(4) Given travel, or opening to steam or exhaust; also both laps, and lead. Strike travel circle and mark points W, Y, and X; diameter BW being known, the steam circle is struck and BA found; and the rest easily completed.

(5) Given steam opening for any particular position of crank, position of crank at cut-off, amount o lead, and exhaust lap. This is answered at (1) Fig. 654. Draw opening 1, 2: lead 1: position of crank for that opening, 3: and position of crank at cut-off, 4. Drop perpendiculars 5 and 6. Draw 7 at 90° to 4, and 8 at 90° to 3. Bisect angle 5, 7, by line 9; and angle 8, 6, by line 10. Their meeting point is the centre of the diagram, the dark line shewing the primary circle.

(6) Given the lead, and the positions of crank at cut-off, release, and compression. See Fig. 654, diagram (2). Let 1 be the lead, while 2, 3, 4 are the positions of crank at cut-off, release and compression respectively. Drop perpendicular 5 and draw 6 at 90° to 2. Bisect 5, 6, by 7, and 4, 3, by 8; their meeting point being the centre of the diagram.

(7) Given lead, maximum opening of steam port, and position of crank at cut-off; also inside lap. For solution see (3) Fig. 654. Let 1 be the lead, 1 2 the greatest steam opening, and 3 the angle of crank at cut-off. Drop the perpendicular 4, and erect 5. Draw 6 at right angles to 3, cutting 4 in A. Bisect 4, 6, by 7, and produce at 8 to G: join 9. Draw 10 horizontally, and with centre A strike 11; join AB by 12. Draw 13 parallel to 12, cutting 7 in E, which is the centre of the diagram.

Zeuner Diagram for Meyer Valve.—Concerning cut-off point only, the real opening to steam will be due to the relation between main and expansion valves at any moment. In Fig. 655, let AB be the stroke of the main valve, CG its steam circle, and θ the angle of advance. Also let θ_1 be the angle of advance of the expansion eccentric (nearly opposite the engine crank), and CH its throw. Taking position E, CF would be the movement of main valve from central position, and CD that of the expansion valve, the difference or relative motion being DF. Measuring this difference at GJ for several positions such as E, CJK is found, which may be proved to be a circle. To find CK directly, join HG and complete the parallelogram HK by parallels

G K, C K. Then, with certain limitations, the radii vector of C K will shew closing to steam of the Meyer valve for respective positions of engine crank.

Let the back valve be adjusted to any desired width, and R be measured when at mid position; with radius R describe the circle L M N O. Strike the steam lap at P R Q; the vectors within P T S U then shew opening to steam for the respective crank positions. Admission is given at P by main valve, in the usual manner; after U the opening is also controlled by the R circle, and when the difference vector equals R, as at C S, cut-off takes place. We see from this diagram how decrease of R secures an early cut-off and *vice versâ*, and rapidity of cut-off can be judged by decrease towards S of the vectors of the shaded area. The exhaust circle is governed by the main valve only. (*See Appendix II.*, *p.* 895.)

Zeuner Diagrams for Stephenson Link Motion.—Knowing that decreased travel when linking-up causes earlier cut-off, we may now examine, by Zeuner diagram, Fig. 656, the exact result obtained. Taking the upper diagram:

Open rods. With throw as radius, strike travel circle F E, and draw valve circle D E for full gear. Draw the link A B, represented by full travel of die, with D A, D B as the distance between die and sheave centres. Through point G, where D A and valve circle intersect, draw E G H to meet the centre line D C in H. Then D H is the diameter of the valve circle in mid gear, and any other circle, as D *e*, will have its diameter bounded by E H; position *e*, between E and H, corresponding to proportionate position between A and C. The centres K L J will form a parabola, lying concave to D and with vertex at J. Draw the lap circle *a b*, and erect perpendicular *d* Y. Y E will be the lead in full gear, and the amount of lead will increase as the travel decreases, shewn by the shaded figure, being *d* H in mid position, or equal to the lap.

Crossed rods. In the lower diagram, the full-gear circle O P is set out as before, also the link M N. The crossing point R, made by the further rod M O, is joined to P, when S P bounds the diameters of valve circles. The centres of the circles now make a parabola convex to O, and with vertex at T. Strike the lap circle *f g*, and draw the perpendicular *g* X. X P is the lead in full

gear, and the shaded portion indicates the change in lead value. Decreasing towards the centre, it vanishes entirely at W, where the opening is h W. At O V the throw is equal to the lap, and therefore the valve does not open at all at positions on the link corresponding to between points V and S.

Space prevents us giving proof of the above, which, while being only approximate, is quite near enough for practical purposes. (*See p.* 1146.)

Ideal Indicator Diagrams for Compound Engines.—We examined in Fig. 622 the form of diagram we should expect to obtain from a single cylinder, and in Fig. 624 some actual diagrams from a three-cylinder compound. The forms in the latter case were sufficiently clear to shew considerable difference of character over those taken from a single-cylinder engine. We shall investigate the ideal diagrams for two-stage compounds, believing that a careful examination will enable the student to carry the method to three or four-stage compounds. To simplify matters, we shall work with numbers instead of letters. Naturally, in building up such diagrams, the only question we ask from time to time is, 'What is the change of pressure with a particular change of volume?' Two formulæ are needed to meet all cases.

(1) When the volume increases or decreases regularly within the same vessel:

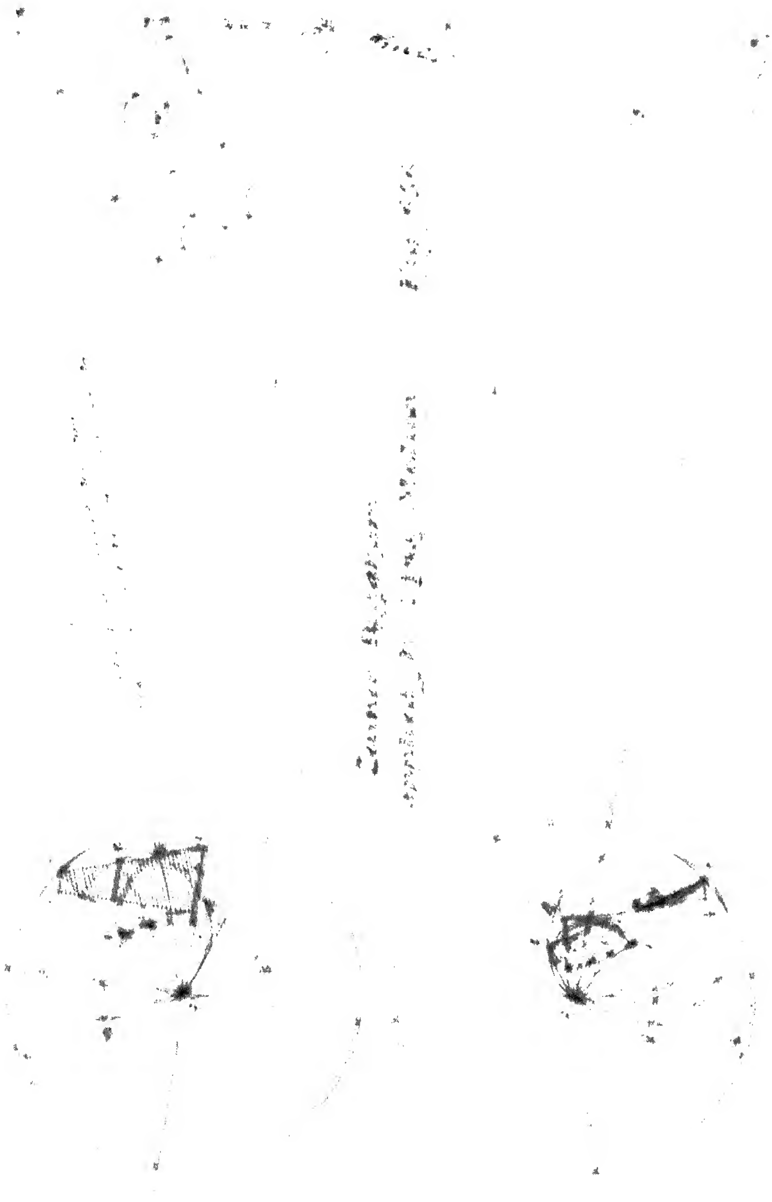
$$p = \frac{\mathrm{P\,V}}{v}$$

(2) When two or more vessels, having each a particular volume, and each containing gas at a particular pressure, are *suddenly* placed in communication:

$$\mathrm{P\ final} = \frac{\mathrm{P\,V} + \mathrm{p\,v} + p\,v}{\mathrm{V} + \mathrm{v} + v}$$

Also, for simplicity, the hyperbola is taken to represent the relation of pressure and volume. See Fig. 620.

I. *Tandem Engine,* with one cylinder behind the other, and both pistons on one rod. Sketching the cylinders at A, Fig. 657, we adopt the artifice of applying a movable paper strip B to



[illegible]

[illegible] H.P. [illegible] L.P. [illegible]

[illegible] H.P. = 4 [illegible] L.P. = 6

Initial steam pressure, 150.

Sometimes a large receiver is an advantage, as in Fig. 629, but, in any case, the receiver must be present, even though only represented by the pipes between the H.P. and L.P. cylinders.

Set up the scale of pressures culminating at 150 [illegible] volumes ⅛ and 1 per clearance, and 1 and 3½ for the [illegible]. Mark off ¼ to P_a, the cut-off in H.P. Draw a hyperbola through P_a to the end of the whole diagram, and measure [illegible] H.P. = 56. Here there is a sudden opening to [illegible], and though we know the volumes of these spaces, we do not know their residual pressures from last stroke. [illegible] these cannot be found directly, but only by trial [illegible] formula [illegible] we recommend that pressures be assumed, and a complete cycle worked in the first place. The residual pressures thus found may be used, a second tour of the cycle made, and so on, till the pressures assumed are equal to those resulted at the end of the cycle. The method seems complicated, but in practice it is not so, for it is never necessary to go round more than three times, and often only twice. On the first stroke assume

Receiver pressure $P_r = 25$
L.P. clearance pressure $P_c = 28$

The latter being fixed by taking a back pressure of 5 lbs., and a compression up to 28 lbs. On the second stroke we measure P_r and find it has risen to 37, P_c remaining, of course, at 28. On the third stroke, by combining P, P_r and P_c we have by formula (?)

$$P = \frac{56 \times 1\frac{1}{8} + 37 \times \frac{1}{2} + 28 \times 1}{1\frac{1}{8} + \frac{1}{2} + 1} = 41$$

ıd now expansion takes place simultaneously in H. P., L. P., d R, up to cut off in L. P. Using formula (1):

$$P_3 = 45 \times \frac{1\frac{1}{8} + \frac{1}{2} + \cdot 3}{\cdot 4 + \frac{1}{8} + \frac{1}{2} + \cdot 3 + 2\cdot 1} = 25\cdot 3$$

ich brings P_3 up to the general hyperbola in the L. P. cylinder. should be noted, however, that only certain proportions between three vessels will do this, and the curve may be either above below the general hyperbola, as in Fig. 659. The rest of the v-pressure diagram, Fig. 657, is easily understood, the compssion curve being a hyperbola through P_L. Intermediate points ween P_2 and P_3 can, of course, be found by calculation.

Following the H. P. diagram, the steam will now be only in nmunication with the receiver, and must therefore be comssed in H. P., H_c, and R, from P_3 to P_4, by formula (1):

$$P_4 = 25\cdot 3 \frac{\cdot 4 + \frac{1}{8} + \frac{1}{2}}{\frac{1}{8} + \frac{1}{2}} = 41\cdot 5$$

Being cut off to exhaust before reaching P_4 we assume a conıient point P_R, and measuring, find $P_R = 37$, or the residual ssure we started with.

II. *Compound Engine with cranks at right angles: cut off in P. after half stroke (at ·6).* This is worked out in Fig. 658, the ult being as follows:

$$\text{Volumes}\begin{cases}\text{H. P.} = 1 \\ \text{L. P.} = 3\frac{1}{2} \\ \text{R.} = \cdot 45\end{cases} \qquad \begin{matrix} H_c = \frac{1}{8} \\ L_c = \cdot 3 \\ \end{matrix}$$

Cut off ... H. P. at ·45, L. P. at ·6.

After first cycle it is found that

$P_R = 17\cdot 7$, and P_L is assumed at 18 as before.

Arriving at $P_1 = 62$, the H. P. piston will be at the end of its ke, but the L. P. piston will be at mid stroke. We therefore e a sudden communication with R, L_c, and half L. P., all ch will have the same residual pressure as R, and

$$P_2 = \frac{62 \times 1\frac{1}{8} + 17\cdot 7\,(\cdot 45 + \cdot 3 + 1\frac{3}{4})}{1\frac{1}{8} + \cdot 45 + 1\frac{3}{4} + \cdot 3} = 31\cdot 4$$

Next the L. P. piston moves to cut-off at P_3, but the corresponding movement of the H. P. piston is so small that it may be neglected. Expanding regularly in all the vessels, we have:

$$P_3 = 31{\cdot}4 \times \frac{1\tfrac{1}{8} + {\cdot}45 + 1\tfrac{3}{4} + {\cdot}3}{1\tfrac{1}{8} + {\cdot}45 + 1\tfrac{3}{4} + {\cdot}3 + ({\cdot}1 \times 3{\cdot}5)} = 28{\cdot}6$$

The rest of the low-pressure diagram up to P_L will be understood. Following the H. P. diagram, P_3 is compressed regularly to P_4 so

$$P_4 = 28{\cdot}6 \times \frac{1\tfrac{1}{8} + {\cdot}45}{\tfrac{1}{2} + \tfrac{1}{8} + {\cdot}45} = 41{\cdot}7$$

And now there is a sudden communication with the clearance L_c, having a residual pressure of 18; therefore,

$$P_5 = \frac{41{\cdot}7 \times (\tfrac{1}{2} + \tfrac{1}{8} + {\cdot}45) + {\cdot}3 \times 18}{\tfrac{1}{2} + \tfrac{1}{8} + {\cdot}45 + {\cdot}3} = 36{\cdot}5$$

Then there is a gradual change of pressure, all three vessels being in communication, but the curve is not a hyperbola, because not only are the cylinders of different area, but the piston speed varies considerably. At centre B strike the larger semi-circle, and at centre A strike the smaller semi-circle, to represent respectively the L. P. and H. P. cranks. Assume the H. P. compression point 6 and join 6 A, then draw B 6 at right angles to 6 A; also divide the portions between 5 and 6 on each crank circle into equal parts, and letter as shewn. Now the total volume at any point between 5 and 6 can be found, it always being $(H_c + L_c + R)$ + vol. in H. P. + vol. in L. P. Thus at P_5, volume $= (\tfrac{1}{8} + {\cdot}3 + {\cdot}45) + {\cdot}5 + 0 = 1{\cdot}375$. For any other position, *w* for example, the volumes in H. P. and L. P. may be found by taking off both the distances * * with dividers and measuring these by the scale F G. We have not space to consider every point, but at P_6 vol. will clearly be ${\cdot}875 + {\cdot}08 + {\cdot}7875 = 1{\cdot}742$. Then,

$$P_6 = 36{\cdot}5 \times \frac{1{\cdot}375}{1{\cdot}742} = 28{\cdot}8$$

Intermediate points between P_5 and P_6 on H. P. diagram being obtained, an arched curve is found as drawn. The H. P. diagram is next completed by drawing a hyperbola through P_6. The L. P. curve from P_5 to P_6 must next be drawn. Now the

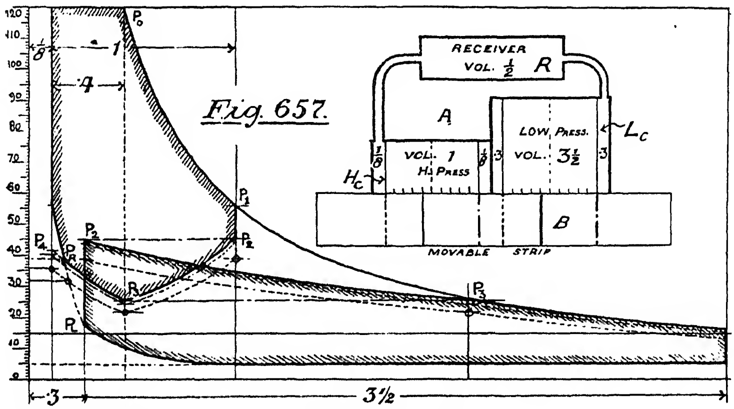


Fig. 657.

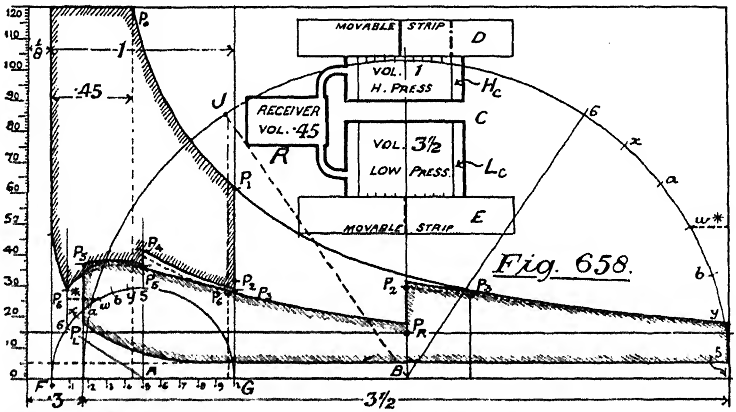


Fig. 658.

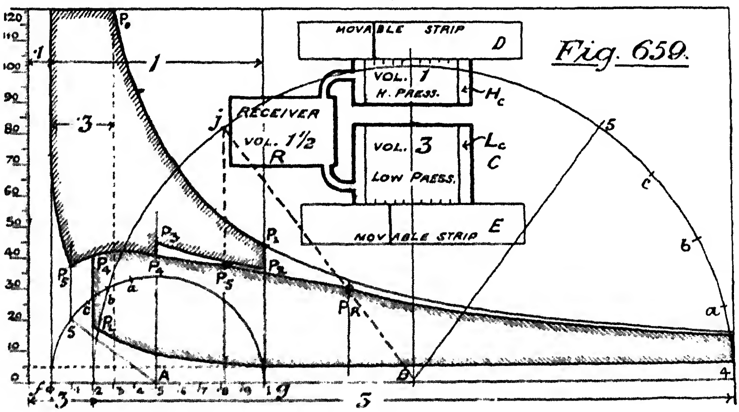


Fig. 659.

pressures have already been obtained for these points, and it only remains to define their volumetric position. To do this take all the points from 5 to 6 on the larger semi-circle, and transfer them to the left side of the circle; thus B 6 is changed to B J. Projecting these downwards we only have to set up the heights previously found, to complete the L. P. curve from P_5 to P_6. The further expansion from P_6 to P_R is only in L. P. cylinder and receiver. Therefore,

$$P_R = 28{\cdot}8 \times \frac{{\cdot}45 + {\cdot}3 + {\cdot}7875}{{\cdot}45 + {\cdot}3 + 1{\cdot}75} = 17{\cdot}7$$

or we have arrived at the residual pressure assumed at first.

III. *Compound Engine with cranks at right angles; cut-off in L. P. before half stroke.* Referring to Fig. 659, and taking the following data:

$$\text{Volumes}\begin{cases}\text{H. P.} = 1 \\ \text{L. P.} = 3 \\ \text{R.} = 1{\cdot}5\end{cases} \qquad \begin{aligned}H_c &= {\cdot}1 \\ L_c &= {\cdot}3\end{aligned}$$

Cut-off ... H. P. at ·3, L. P. at ·4.

P_R will be found to be 30·2, while P_L is 18 as before. $P_1 = 44$ by measurement. Then the drop to P_2 is much less than that in Fig. 658 because the receiver *only* is opened to H. P. cylinder, and

$$P_2 = \frac{44 \times 1{\cdot}1 + 30{\cdot}5 \times 1{\cdot}5}{1{\cdot}1 + 1{\cdot}5} = 36{\cdot}2$$

Compressing in H. P. and R,

$$P_3 = 36{\cdot}2 \times \frac{1{\cdot}1 + 1{\cdot}5}{1{\cdot}1 + 1{\cdot}5 - \frac{1}{2}} = 44{\cdot}8$$

A sudden expansion occurs by opening to L_c, and

$$P_4 = \frac{44{\cdot}8 \times \{\frac{1}{2} + {\cdot}1 + 1\frac{1}{2}\} + 18 \times {\cdot}3}{\frac{1}{2} + {\cdot}1 + 1\frac{1}{2} + {\cdot}3} = 41{\cdot}4$$

Then, while L. P. crank moves from 4 to 5 on the large circle, the H. P. crank moves through a similar arc on the smaller circle, at right angles to it, as before. Taking volumes at P_4 and P_5 we have

$$P_5 = \frac{41{\cdot}4 \times \{{\cdot}5 + {\cdot}1 + 1{\cdot}5 + {\cdot}3\}}{{\cdot}1 + {\cdot}1 + 1{\cdot}5 + {\cdot}3 + {\cdot}61} = 38$$

Finally, expanding from P_5 to P_R in L. P. cylinder and receiver,

$$P_R = 38 \times \frac{1 \cdot 5 + \cdot 3 + \cdot 61}{1 \cdot 5 + \cdot 3 + 1 \cdot 2} = 30 \cdot 5$$

the residual pressure.

While a small receiver should be adopted in Case II., a very large one is advisable in Case III. in order to equalise the work in the H. P. and L. P. diagrams. Of course, Case II. compels a large gap in the combined diagram, on account of drop in receiver and low-pressure cylinder, and the arrangement is not, therefore, counselled. The student should compare actual diagrams with ideal ones, and endeavour to distinguish between Cases I. and III.

Correction of Indicator Diagram for Inertia.—The indicator diagram, as obtained from the cylinder, does no more than transcribe the changing pressure and volume on one or other side of the piston. The actual pressures tending to move the piston are not correctly shewn, at least not without a small correction; but those transmitted to the crank, which are what we most require to know, are considerably different, on account of the deductions and additions required to respectively start and stop the reciprocating parts at the beginning and end of each stroke. We shall now examine the modifications to be made in the indicator diagram in order to arrive at the tangential pressure on the crank pin; and, to make the investigation as useful as possible, shall take an actual case of a vertical engine, where there is not only the inertia force to contend with, but the dead weight of the moving mass. In a horizontal engine there is no such dead weight, while in a diagonal engine the pressure along the incline caused by the weight is the effective resistance.

Let the crank circle, J K L M, Fig. 660, have a radius of 1′ 9″, as measured by its own scale. Divide the circumference into, say, 20 equal parts, and, with a connecting rod 7′ 6″ long, mark corresponding positions of piston stroke from A to B. Draw the polar curves, K U and U M, by the method given at p. 491, and transfer the ordinates to the base A B, so as to form the velocity

curve AXB. Supposing the crank to revolve uniformly at eighty-eight revolutions per minute, the velocity of crank pin,

$$v = 16{\cdot}1 \text{ ft. per sec.}$$

that is, XY should measure 16·1. Dividing this ordinate into 16·1 parts will give the scale of piston velocity.

Next find the acceleration curve, QTR, adopting the method already explained at p. 492, and illustrated in Fig. 454. QT will shew, from base AB, the rate of *increase* of velocity, and TR the rate of *decrease*, for the top diagram, viz., when the crank moves through JKL; but on the return stroke, from B to A, lower diagram, RT will be acceleration and TQ retardation. The acceleration scale will not be the same as the velocity scale, but must be compressed in the ratio $\frac{d}{v}$ as explained on p. 492. In other words:

Reading on acceleration scale

$$= \text{reading on velocity scale} \times \frac{v}{d},$$

Produce X horizontally to Q. Then

$$\text{Reading AQ} = \frac{16{\cdot}1 \times 16{\cdot}1}{1{\cdot}75} = 148$$

By dividing AQ into 148 parts, an acceleration scale is there-ore formed. (*See pp.* 932 *and* 1107.)

Now the force required to produce a given acceleration in a given mass (p. 473) is $\frac{w}{g}f$: that is, the inertia force is proportional to the acceleration. The weight of moving parts in this engine is 8030 lbs., and the inertia force at any moment,

$$F = \frac{wf}{g} = \frac{8030}{32{\cdot}2} \times \text{acceleration reading.}$$

The acceleration curve may then be transformed into a *curve of inertia pressure* (*total*) by multiplying by the above fraction or by $8030 \div 32{\cdot}2 = 249{\cdot}4$, that is, the distance AQ must be divided into $148 \times 249{\cdot}4 = 36{,}911$ parts. This has been done along BP.

From the total pressure scale take 8030 lbs., with dividers, and move the curve QR down by that amount, to NP, thus repre-senting the dead weight of the reciprocating parts.

It is convenient to make one more scale, to show *pressure per square inch* of piston. The piston area being 491 square ins., divide the total pressure reading by 491 to obtain reading per sq. in.; stepped off at s*t*.

The indicator card for the top of the piston is set out by the unit pressure scale at s*t*, and appears as EQXHB, the bottom of diagram touching the base AB. Similarly FPGA is the card from the bottom of the piston. Now, while QXHB is being drawn by the indicator on top side of piston, AFR would be formed by that connected with the bottom side, and the effective pressure will be the difference of these curve ordinates. Deduct those at F from those at H, and the result is the curve WR. So also VN is the curve of effective pressure on the bottom side of the piston.

Now the actual total pressure to be carried forward to the crank pin will be, during the first half of the stroke, less than that on the indicator diagram by the amount required to set in motion the reciprocating masses, viz., their inertia; and during the second half of stroke the indicated pressure will be increased by the backward pull needed to absorb inertia. Briefly, then, the 'top' card loses by the area ANS, and gains by SBP, the resulting pressure area being NXWP; and similarly the resulting area for the 'bottom' card will be P*t*VN. Setting up the resulting ordinates on the straight base AB, we have the curve A*bd*B for the top and B*ef*A for the bottom of piston, the *total pressures* being written on each ordinate; and in order to equalise the areas the cut-off in top diagram has been placed at ·3 and in bottom at ·6 of stroke, the dead weights having to be supported in the latter case.

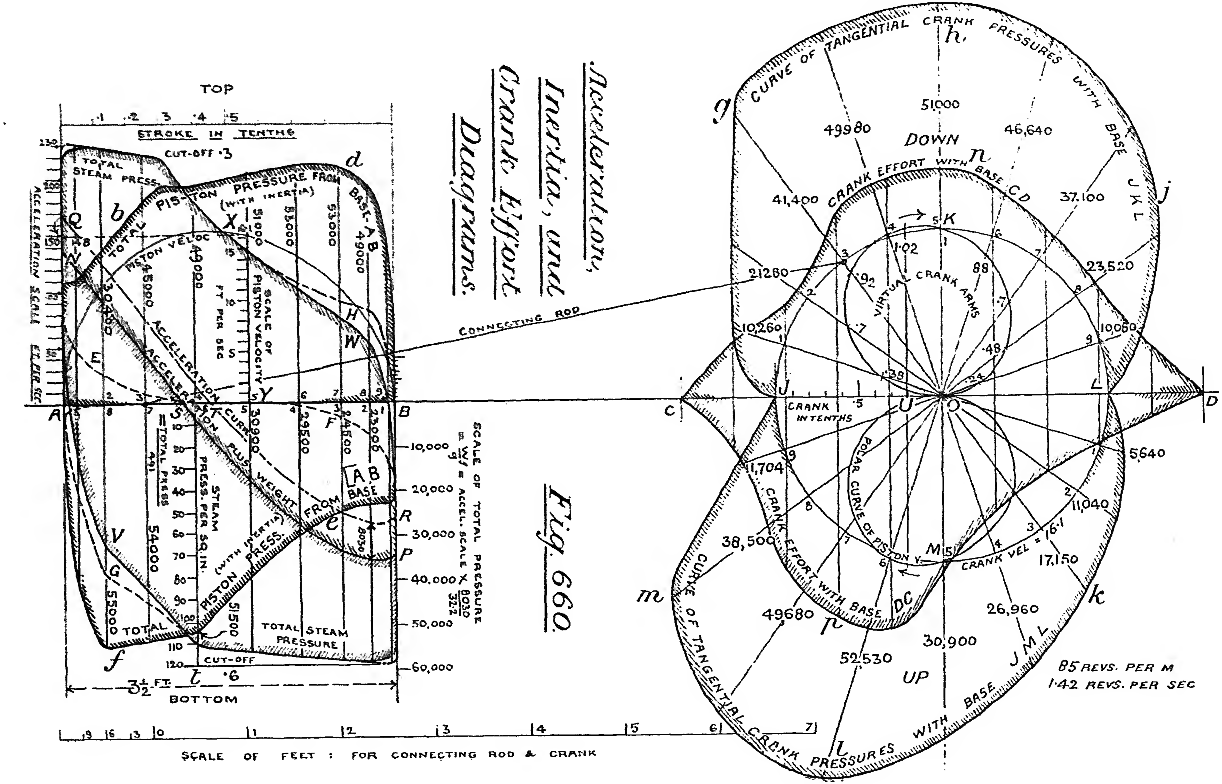
We must next distinguish between reciprocating and rotating parts, for only the former cause inertia force. The piston, piston rod, crosshead, and smaller end of connecting rod are reciprocating weights, but the larger end of connecting rod is a rotating weight. As regards the connecting rod itself, about two-thirds may be called reciprocating and the remaining third reckoned as a rotative weight. The reciprocating weights directly affect the indicator diagram, and the latter must be altered, by increased compression or later cut-off, until a fairly even pressure is

obtained. The revolving parts must be balanced by opposing weights on the crank shaft.

Curves of Crank Effort.—If the crank be on either dead centre, there is no tangential or turning effect produced by the steam pressure on the piston, all such pressure being received upon the bearings. When the crank is midway between dead points the whole piston pressure is transmitted tangentially, and there is no pressure on the bearings except that due to dead weight. Between the two conditions part of the pressure is transmitted tangentially and part normally. But (p. 491) the polar curve proportionally represents tangential crank pressures, other things being equal. Divide then JO, Fig. 660, into tenths and measure the radii vector of the curve UK in terms of these divisions: the numbers obtained will represent the virtual crank arms in relation to pressures transmitted along ABO. Taking the total pressures from A to B, multiply each pressure ordinate by its virtual crank arm, and the result will be the tangential crank pressure for that position. Setting out these results radially, with the crank circle JKLM as a base line, we obtain the two curves of crank effort J*ghj*L and L*klm*J for the top and bottom of piston respectively. These again are better understood on a straight base, so the base JK is stepped out at CO, KL at OD, and the radial ordinates transferred as vertical ordinates on the new base CD. Curves C*n*D and D*p*C are thus arrived at.

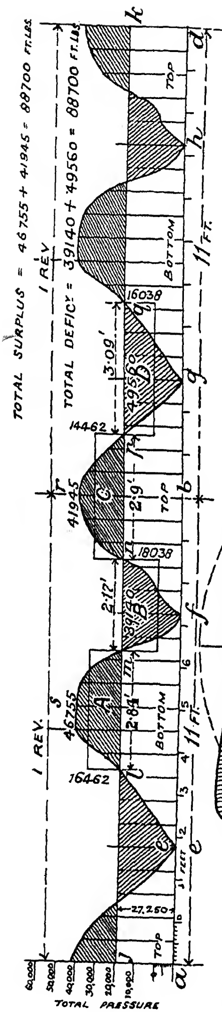
Combination of Crank Effort Diagrams.—Though the fly wheel may equalise very tolerably the crank effort, there is still the difficulty of starting when the crank is at either dead centre. This is not a material difficulty for a factory engine which has only to be started twice a day; but in locomotive and marine practice it would be a very serious obstacle. In locomotives two cranks at right angles are employed, as at 1*a*, Fig. 447, p. 486, while in marine practice it is usual to place three cranks at 120° mutually. The latter gives the best conditions, but the advantage of both will be made clear in Fig. 661.

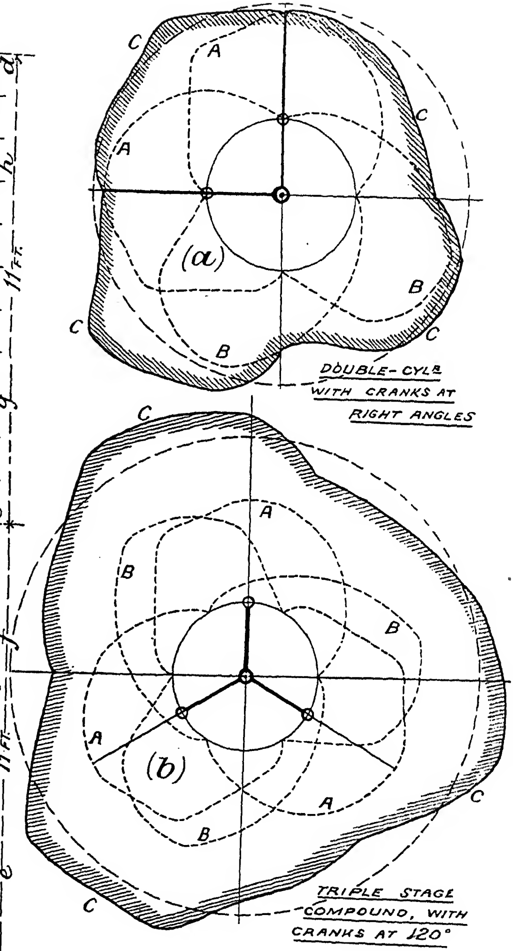
At (*a*) two cranks set at 90° are each supposed to have effort curves, as in Fig. 660. Plot these with relation to the respective cranks, AA being the top curves, and BB the bottom ones. Then the curve of total effort may be found by super-position, that is, at



Acceleration, Inertia, and Crank Effort Diagrams.

Fig. 660.





Combination of Crank-Effort Diagrams
Fig. 661.

Fluctuation of Crank Energy
Fig. 662.

every radius the ordinates of both curves are added to form the resulting curve C C. An average circle is struck, and shewn dotted; and a clear conception of the more even turning movement is then obtained.

Three cranks are set out at (*b*), Fig. 661, and the like process followed. The same letters are adopted throughout, and a more regular turning movement results. The differences between the C C curve and the dotted circle may seem little better than before, but they form a much smaller *percentage* of the effort ordinate.

Calculation of Fly-wheel Weight required.—The crank radius, Fig. 660, being $1\frac{3}{4}$ feet, the circumference of the crank circle is exactly 11 feet. In Fig. 662, let *a d* be 22 feet, and let it be divided so that *a e* and *h d* are each $2\frac{3}{4}$ feet, and *e f*, *f g*, and *g h*, are each $5\frac{1}{2}$ feet. On *e f* and *g h* set up ordinates of crank effort on the up stroke, and on *f g* of that on the down stroke, those on *a e* and *h d* each representing half the down stroke effort. Now take the mean of the ordinates on *e f*: dividing the base into 10 parts, measuring at *centre* of each part, adding the ordinates and dividing by 10: the result is 29,500 lbs. The mean of the ordinates on *f g* is found similarly to be 25,000 lbs. Adding and dividing by 2, gives 27,250, the mean effort for the continuous diagram *a d*. Draw *j k* at this pressure above *a d*.

Now the areas, A, C, &c., shew surplus work, while the crank travels from *l* to *m*, and from *n* to *p* respectively, while the areas B, D, &c., shew a work deficit between *m n* and *p q*. The fly wheel must absorb the work A or C, and give it out again at B or D, and thus tend to equalise the crank effort. The mean pressures and distances traversed have been measured at A, B, C, and D, and are shewn by work rectangles. The total surplus and the total deficit of work per revolution are each found to be 88,700 foot pounds, and the greatest of the four work areas A, B, C, and D, is D, or 49,560 foot pounds. This is the amount of energy which the fly wheel must be able to deliver, such delivery decreasing its velocity, while the absorption of energy will in like manner increase it. But the heavier the fly wheel, the less will be the fluctuation of velocity; and the problem is to find the weight of wheel which will absorb the surplus energy and re-deliver it

keeping the fluctuation of velocity within a certain percentage of the mean. Let v = mean velocity, and let

$$\text{Total fluctuation of velocity} = \frac{1}{k} \text{ of } v$$

then the value of k depends on the regularity required, and may vary from 100 for very steady driving, to 20 where constant speed is of little value. With feet and seconds units, let v_1 be maximum velocity and v_2 minimum velocity of the fly wheel at its mean radius, consequent on absorbing and delivering the given energy, and let E *represent the energy area*, or the 49,560 foot pounds of Fig. 662, while the velocity falls from v_1 to v_2.

$$\text{Energy delivered} = \frac{w\,(v_1^2 - v_2^2)}{2\,g}$$

where w is the weight of the fly wheel. But this energy is equal to the area E,

$$\therefore\ \mathrm{E} = \frac{w\,(v_1^2 - v_2^2)}{2\,g}$$

$$\text{Now } v_1 - v_2 = v \times \frac{1}{k} \qquad v_1 + v_2 = 2\,v$$

$$\text{and} \quad v = \frac{2\,\pi\,\mathrm{R\,N}}{60}$$

R being radius of gyration of fly wheel.

Putting also the fly-wheel weight in tons,

$$\underline{\mathrm{W}} = \frac{\mathrm{E} \times 2\,g}{(v_1 + v_2)\,(v_1 - v_2)\,2240} = \frac{2\,\mathrm{E}\,g\,k}{2\,v^2 \times 2240}$$

$$= \frac{\mathrm{E}\,32{\cdot}2\,k\ \ 60 \times 60}{4\,\pi^2\,\mathrm{R}^2\,\mathrm{N}^2\,2240} = \underline{1{\cdot}31\,\frac{\mathrm{E}\,k}{\mathrm{R}^2\,\mathrm{N}^2}}$$

Generally R may be taken at centre of rim section, but if great accuracy be required, assume a fly-wheel section, replacing the arms by a thin disc of equal weight: then, moment of inertia of volume (second moment) of a solid of revolution, or $\mathrm{I_v}$ = volume $\times\,\mathrm{R}^2$, which is the third moment of the generating

area (A), or of half the fly-wheel section. But volume = first moment of A

$$\therefore\ R^2 = \frac{I_v}{\text{vol.}} = \frac{\text{3rd moment of A}}{\text{1st moment of A}}$$

the moments being found graphically at p. 845. If inches have been adopted, R must be changed to feet when inserting in the fly-wheel formula. Also v must be measured at radius R. If W does not now agree with the calculated weight, the section must be altered, and a new calculation made.

Area of Steam Port.—Practice has decided certain average speeds of piston in particular cases, and the following list has been thus deduced:—

MEAN PISTON SPEED IN FEET PER MINUTE.

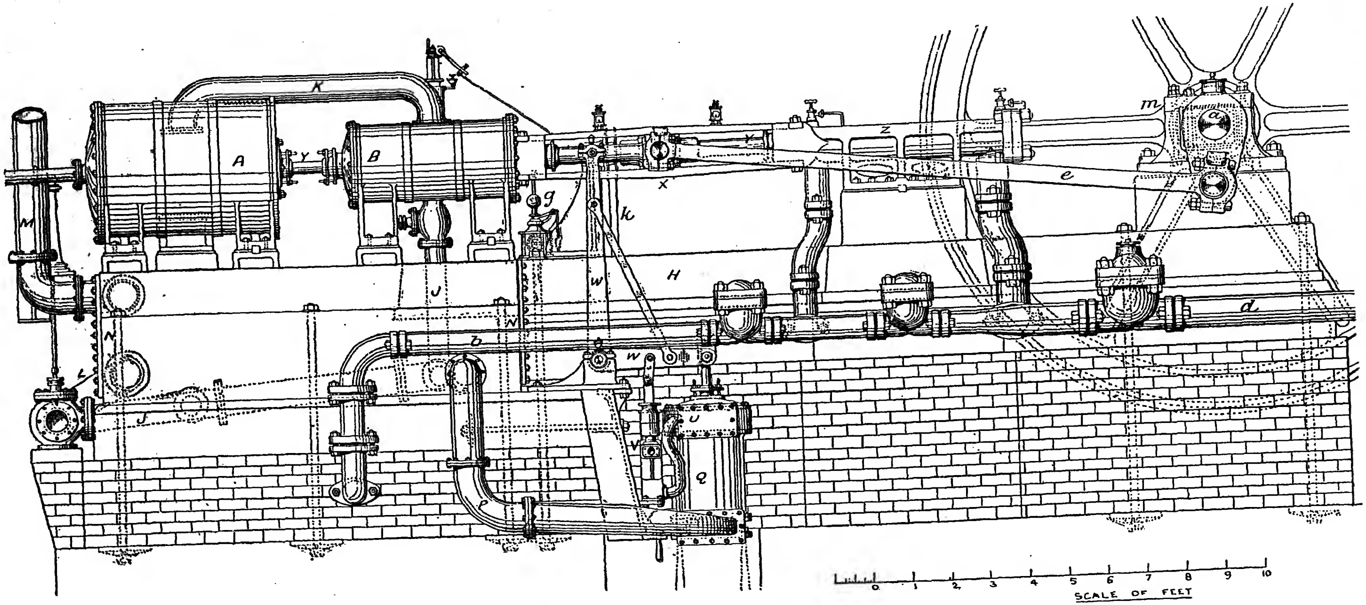
Locomotives	1000
Marine engines	700
Horizontal engines	400
Pumping engines	130

To attain this speed, we must not endeavour to pass the steam through the steam port at a greater speed (according to Rankine) than 100 feet per second.

$$\therefore\ \text{Ratio of } \frac{\text{cylinder area}}{\text{port area}} = \frac{100}{\text{speed of piston in ft. per sec.}}$$

If the port be much contracted, a lower piston speed will be attained than that intended. We shall close this chapter with some practical examples, together with a few remaining points of theory thereupon raised.

Horizontal Compound Pumping Engines.—Figs. 663-4-5 are views of a pair of compound engines designed and built by the East Ferry Road Engineering Works Company, Millwall, serving as examples of the Stationary or Land engine. The engines are used at the Millwall Docks for pumping water to hydraulic accumulators, under a pressure of 750 lbs. per sq. in. The bed plate H is in two parts, and supports the high-pressure cylinders BB, the low-pressure cylinders AA, the pump cylinders ZZ, and the crank shaft bearings *m m*. The H. P. cylinder is supplied with a main valve D, and an expansion valve E, of Meyer form, the valve spindles being lettered respectively G and F. The



Horizontal Compound Pumping Engines.

(BY THE EAST FERRY ROAD ENGINEERING WORKS Co)

Fig. 663.

(Reduced by permission from "Engineering")

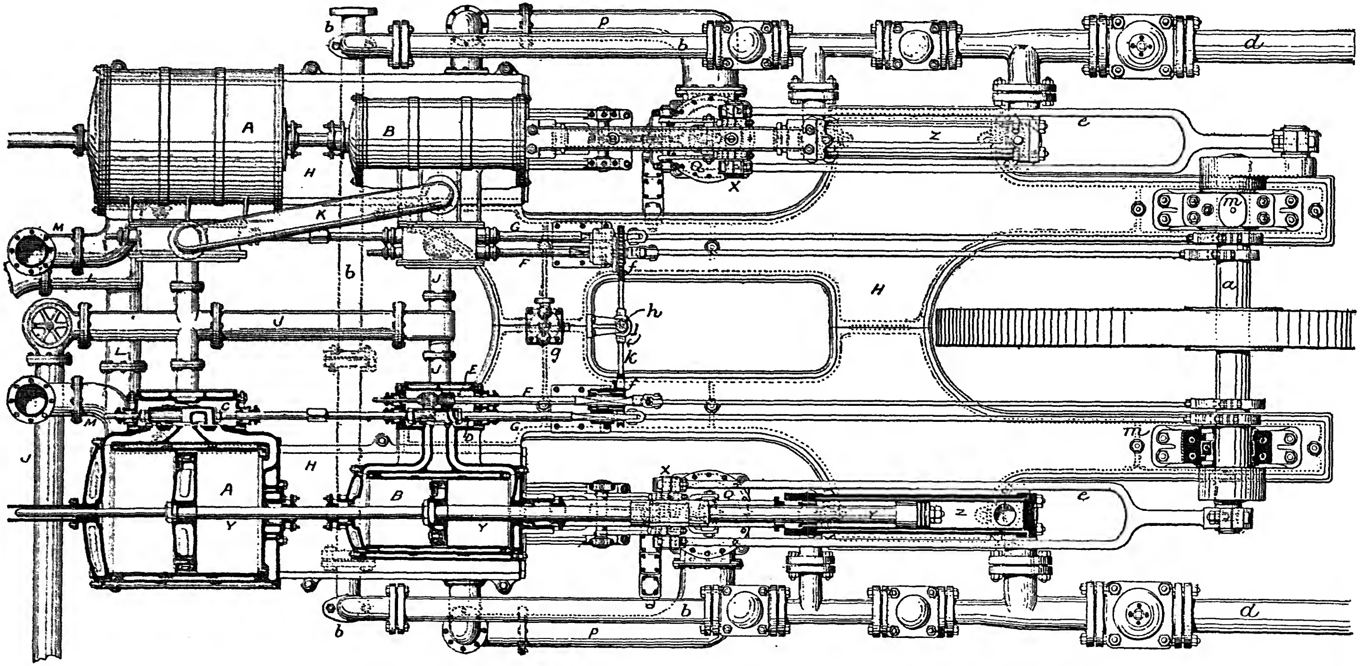


Fig. 664.

valve C for the L. P. cylinder is double-ported, but not relieved from steam pressure at the back. The piston rod Y Y, being prolonged, forms the pump plunger, its sectional area being half that of the pump piston, for reasons to be explained in the next chapter. J J is the steam supply pipe to the H. P. cylinder, K its exhaust, as well as supply for L. P. cylinder, and the L. P. cylinder exhausts directly into the condenser, as will be seen in Fig. 665.

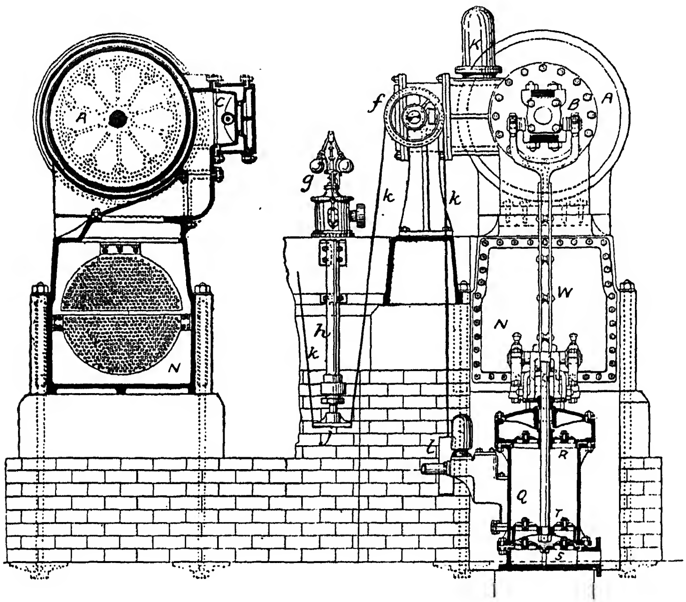


Fig 665.

The surface condenser N N consists of a rectangular casing, containing a nest of tubes. Cold water being allowed to flow in through the pipe M, passes through these small tubes, first through the top half, returning through the lower half, then out by the pipe L. The exhaust steam distributing itself outside the tubes, becomes condensed, is taken away as water by the air pump Q, and delivered to the hot well U; then by the feed pump V to the boiler. The air pump bucket has four valves at T, fixed foot

valves S, and delivery valves R to prevent the water returning. The bucket is actuated by the bell-crank lever WW, connected by links to the crosshead X. The connecting rod *e* has a long fork to clear the pump barrel; it is also light in construction, its sole duty being to transmit equalising energy to or from the fly wheel, in addition to the power required to work the valves. The pump suction pipe is at *d d*, and the delivery pipe at *b b*, but full explanation will be left to the next chapter. We must not omit to mention the hydraulic governor *g*, the invention of Mr. C. R. Parkes, M.I.C.E., which has given great satisfaction in its working. The flying balls are driven from the engine in the usual manner, but the sleeve opens a small D valve to hydraulic pressure or exhaust, according to whether it rises or falls. Nothing takes place until the governor has attained a speed of 15 revolutions per minute, when high-pressure water is admitted into the cylinder *h*, and the ram *j* is pushed downward, thus also pulling down the strap *k* and raising the weight *l*. The consequence is that the pulley *f*, on the expansion valve spindle, is rotated so as to increase the lap of the Meyer valve and secure an earlier cut-off, and the action will continue until the speed of the engine has returned to the normal, when the governor sleeve will fall, open the D valve to exhaust, and allow the weight *l* to lift ram *j* to its original position.

Triple Expansion Marine Engines.—Figs. 666 and 667, Plate XVII., are two views of the triple-expansion engines of the Pacific steamer *Iberia*, designed and constructed by Messrs. David Rollo and Sons, of Liverpool. The *bed plate y*, in three pieces, carries the left-hand standards; the right-hand standards K, K_1, and K_{11}, being built upon the condenser V. *Cylinders.*—The H. P. cylinder A is 33 ins. diameter, B the intermediate is 54 ins., and C the L. P. cylinder is 88 ins.; while G, H, and J are the respective *pistons*, of conical form to combine lightness with strength, and each having a stroke of 60 ins. To minimise the number of spare parts, the *cranks* YYY, *connecting rods* ZZZ, *piston rods* DEF, *eccentric* S *and rods* STU, *links r*, *gudgeons zz*, *crossheads v*, and *pump levers jk*, are all made respectively interchangeable; only a small alteration occurring with the rod D, which must have the tail or upper part cut off. *Valves.*—A piston valve *b* is

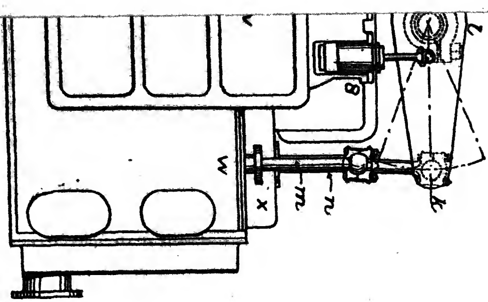
adopted for the H. P. cylinder, packed with flat rings, but distributing steam like a D valve. To save space two piston valves *c* are supplied to the I. P. cylinder, as seen by the valve rods 3 and 4, connected at their lower ends by the strong crosshead 5; and the L. P. valve *d* is double-ported, while being relieved on its back by the hollow piston *h*. Piston *g*, with steam pressure underneath, supports the weight of valve *d*, and the I. P. valves are similarly supported by pistons within *x* and *w*. *Relief valves uu*, on the cylinder covers, are wing valves weighted with springs, serving as safety outlets for condensation water, which might otherwise break the covers when the pistons moved. The H. P. and I. P. slide valves, being vertically above the crank shaft, their eccentrics are set to lead the cranks, but the L. P. valve is moved by the rocking lever Q R, and its eccentrics must therefore follow their crank (see p. 646). The *radius link* is formed of two plates, having the die between and the eccentric rods outside, thus enabling the pin centres to be coincident when in full gear. The steam reversing cylinder *t* has its rod *u* coupled to the weigh bar lever *sp*, which, through the drag link *qr*, moves the link *r* to fore or aft positions; expansive adjustment is given by the screw *q*, and 9 is the valve lever for cylinder *t*. The exhaust steam passes to condenser V through standard K; and air pump X and circulating pump W are worked from crosshead *v* by levers *jlk*. 7 and 8 are oil pumps, and 6 an oil reservoir with gauge. The cylinders are jacketed at sides, top, and bottom; and drains connect to tanks which shew the water used.

Condensers.—The advantages of condensation having been discussed theoretically, we will now describe the three principal methods of realising those advantages practically.

The *Jet Condenser*, Fig. 668, applied to most land engines, consists of the condenser A, where exhaust steam E is met by a constant spray of cold water from injection cock G; the air pump B, worked from the engine piston rod; and the hot well C, from which the condensed steam and water is taken to feed the boiler. In order to make B's action continuous, there is a suction valve S and a delivery valve D at each end of the cylinder.

The *Surface Condenser*, Fig. 669, avoids the mixing of cooling water with the steam directly. Formerly, if such water were dirty

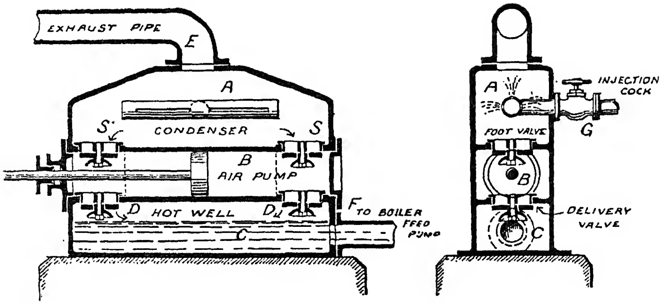
PLATE XVII.



Triple-expansion Engines of the Pacific Steamer "Iberia."

(BY MESSRS. DAVID ROLLO AND SONS.)

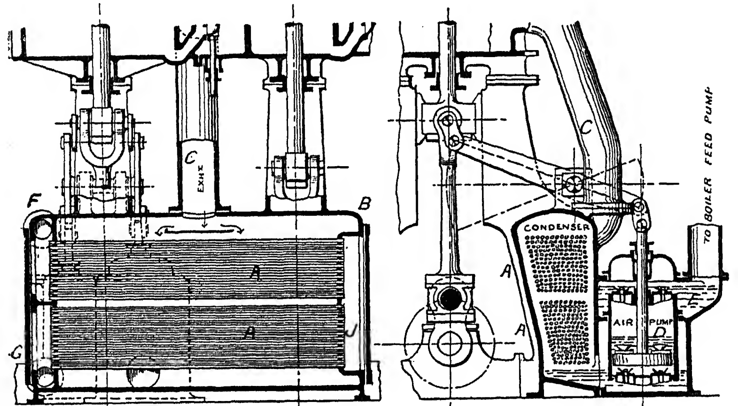
(Reduced by permission from "Engineering")



Jet Condenser

APPLIED TO HORIZONTAL ENGINE

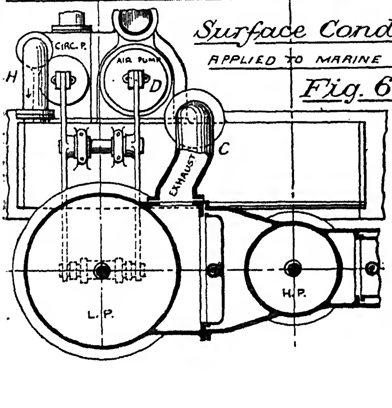
Fig. 668



Surface Condenser

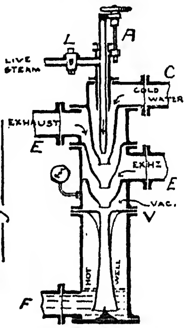
APPLIED TO MARINE ENGINE

Fig. 669.



Morton's Ejector Condenser

Fig. 670.



[illegible]

[illegible] plates and the [illegible]

A large [illegible] the exhaust steam [illegible] condensed [illegible] being [illegible] the air pump [illegible] the hot well [illegible] circulating pump [illegible] takes condensing water from the sea [illegible] forces it [illegible] through the [illegible] tubes [illegible] along the [illegible] and out to the sea again at [illegible] considerable heat [illegible] thrown away, but the method is more economical than that of intermittent blowing off of hot water from the boiler.

The *[illegible] Condenser* [illegible] is there arranged [illegible] cylinder engine [illegible] are the exhaust pipes, and [illegible] to a cold water tank having, preferably, a little [illegible] introduced [illegible] starts the action [illegible] after [illegible] condensing water adjusted [illegible] so rapidly as to cause a vacuum [illegible] from the hot well [illegible] as before [illegible]

Further Marine Details. [illegible] details already described, there are [illegible] the propeller, and the propeller shaft.

The *Screw Propeller* [illegible] blades [illegible] bolted separately to the boss [illegible] The boss is cast steel [illegible] is a section, and [illegible] a development of the [illegible]

Fig. [illegible] represents the position [illegible] and propeller, the length of shafting [illegible] position, which is usually [illegible]

The *Thrust Bearing* [illegible] as applied to the *[illegible]* [illegible] the surface resisting propeller thrust, there are [illegible] bearings [illegible] and [illegible] corresponding collars [illegible] on the shaft. The bearings [illegible] with white metal, and supplied with oil [illegible] are [illegible] screwed bolts [illegible] fixed to the main casting [illegible] and [illegible] adjust the bearings, that each takes its proper share of the [illegible] [illegible] are ordinary supporting bearings with oil [illegible] The shaft is cooled by circulating water beneath it, and the [illegible]

horseshoe bearings have water passing in and out at J J. K K are lifting eyes.

The *Stern Tube*, Fig. 673. A is the tail shaft, tapered to fit the propeller, where it is keyed and gripped by a nut and split cotter. A renewable muntz-metal sheathing D is rolled on the shaft, and gives a smooth, non-corrosive working surface. The tube B, bolted to the water-tight bulkheads at H H, and supported by the stern frame at C, has a bush E in which are placed staves of hard wood (*lignum vitæ*), being the best bearing where water is the lubricant. At the other end a stuffing-box, formed by the neck ring F and gland G, prevents water entering the tube.

Compound Locomotive.—The general arrangement of a locomotive being well known, one good typical example will here suffice. The example chosen serves to illustrate the ordinary 'inside cylinder' engine, having cylinders within the frame, the only main difference being the arrangement of steam pipes. It also shews one of the most successful adaptations of the compound principle to locomotives.

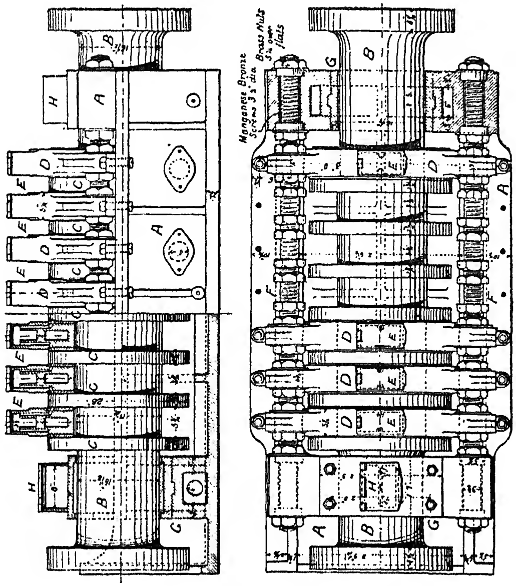
Figs. 675–6–7, Plate XVIII., are views of a Compound Express Locomotive for the North-Eastern Railway, on the 'Worsdell and Von Borrie' principle. The *main frame* consists of two plates L L, a cross stay L_1, and *buffer beams* M M: the front beam carrying the *buffers* N N, *draw hook b*, and *coupling screw c*, while the back beam faces that of the tender Q. Between M and Q are placed buffers P, pivot 50, and safety links 88, the pull being taken by the *draw-bar* 6. 35 is the *foot-plate*, 19 the *cab*, to shield from the weather, 34 the *platform*, and *y* the *splasher* for the driving wheel: *f f* are lamp brackets, and *d d* lifeguards. The *cylinders* A and B are bolted between the frame plates, and *slide valves* $a\,a_1$ are placed above the cylinders to suit Joy gear, whose various links Z, Y, X, and W are explained at Fig. 640. There are four *slide bars q q* to each cylinder, and two *motion blocks r r*: *n* and *p* are the *piston rods*, and *m m* the *connecting rods*. The *weigh-bar shaft s* is moved by a hand-wheel and screw at *v*, coupled to lever *t* by the rod *u*. E E are the *driving wheels*, and F F the *trailing wheels*, with J and K the respective axles: the former is known as the *crank axle*, and in the N.E.R. example is turned throughout. The wheel centres are of cast steel, but the

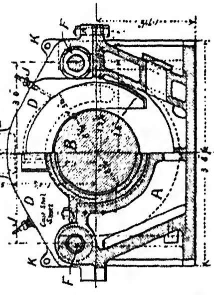
tyres are rolled weldless and fit into annular grooves in the wheel rim, to resist centrifugal force. The front end of the frame is supported by a trolley or *bogie*, which permits certain side movement when travelling round curves. H H is the bogie frame, with stay rods TT, CC the bogie wheels, and GG the axles. A block or die 43, curved to a radius from 42, is held by the pin D; and guides 44, similarly curved and forming part of the bogie frame, ride upon the die. If 43 were rigid, the bogie would only swivel round 42, and would only adjust itself to certain curves; but the freedom of 43 on D permits a further angular movement, and the virtual centre 42 is therefore variable. The buffers U U limit the lateral deviation, and springs *g g* return the bogie frame to central position.

Laminated springs, *h* and 13, and helical springs, 45, placed between frames and wheels, lessen the shock due to inequality of permanent way, and the necessary vertical sliding is met by providing special bearings, S, *x*, and 14, termed *axle boxes*, for the bogie, driving, and trailing axles respectively. R, *w*, and 15 are the *guides* in which the boxes rise, and *z* is a wedge to take up wear in the main box.

A hand brake 11 is used in emergency, but the regular work is performed by the *Westinghouse compressed air brake.* The steam pump 51 fills the main air receiver 5, from which auxiliary receivers—one to each carriage, and one, 46, for the engine—are further supplied. From 46 pipes are led to the cylinders 4 4, and the air pressure moving levers 2 2 put the blocks 3 3 on the tyres. Upon exhausting, 3 3 are released by springs. When ascending steep gradients, sand is driven between wheel E and the rail by means of a steam jet from pipe *l*, the sand passing from *sandboxes j j*, down the pipe *k*. *Cylinder cocks* 18 act as relief valves, and are opened after the engine has stood some time.

Boiler 20, and firebox 21. Very little description is needed beyond that at p. 335. Girder stays 53 are of cast steel, and 52 are long stay bolts. A firebrick arch 23 deflects the current of heated gases over the box, and the *ash pan* 2 has doors or *dampers*, both before and behind, for regulating the draught. The draught is 'induced' by the exhaust steam escaping at the





THRUST BLOCK

Fig. 671.

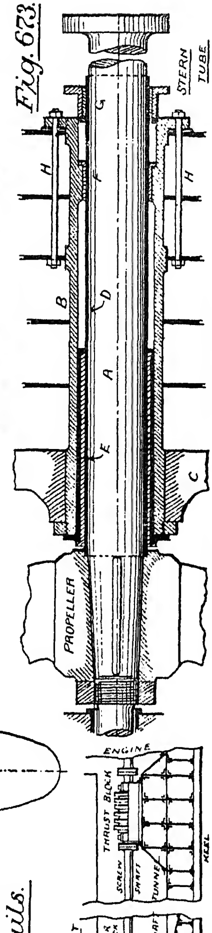
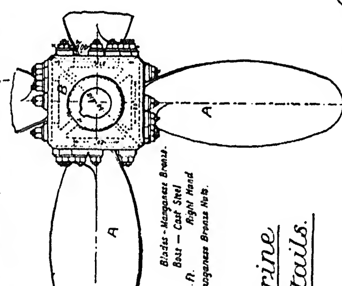
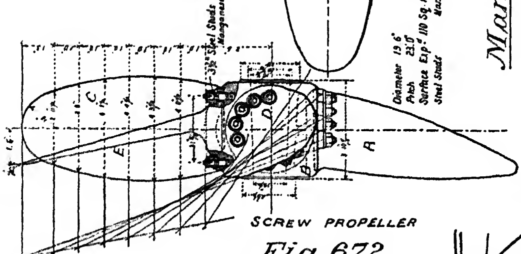


Fig. 673.

SCREW PROPELLER

Fig. 672.

Marine details.

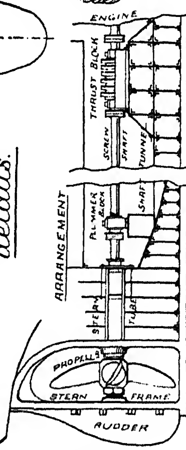


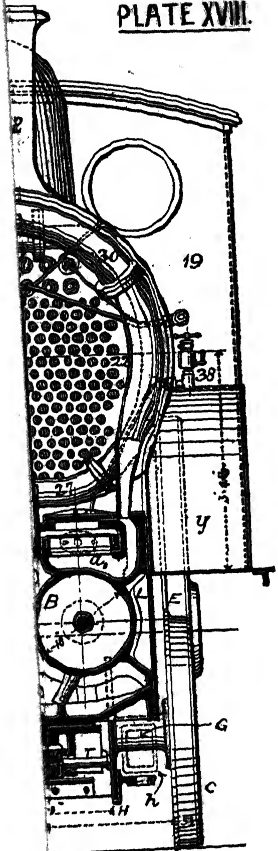
Fig. 674.

blast pipe 39, gradually contracted towards the orifice to cause the necessary velocity; and the *smoke box* 22 must be air tight, so its door 33 is provided with two handles, one for turning the tongue catch, and the other for tightening the screw. A jet of steam from the *blower* 40 causes draught when the engine is standing. The *steam regulator*, 56, has two slide valves worked from handle 34, the main valve 27 being treble-ported, and the 'easing' valve 28 double-ported and small. A pin 55 connects both valves to the gear; but the hole in 27 is slotted, so that when opening, 28 is first moved (easily, being small) and a film of steam admitted between the main valve and its seat. Next, 27 is caught by the pin, and, on account of the relief just given, can be moved without difficulty.

Under ordinary conditions steam first enters the H. P. cylinder B by the pipe 27, exhausts thence to the L. P. cylinder through 30 and 28 (the whole pipe forming a receiver of a capacity equal to B), and finally leaves by the blast pipe 39. But if H. P. crank be on a dead centre at starting, steam must first be admitted to the L. P. cylinder A, and yet be prevented from entering B for fear of blocking the piston. Outside the smoke box a valve box 61 is fixed, having a *starting valve* 59 opened by a rod from the foot plate when required, but at other times kept closed by a strong spring. Pipe 41 takes steam from the boiler to 61, and 57 carries it away to the main pipe 28, entering at 29; and a piston 62, fitting in the valve box 61, is connected to the rod 60 for the purpose of lifting the *flap* or *intercepting valve* 58, which is normally open. When the driver wishes to start, he opens regulator valve 27, and if the H. P. piston refuses to move, he pulls the small lever which opens 59; and steam, wire-drawn to half pressure, enters 61, moves 62 clear to the left and closes 58, then passes by 57 and 29 to move the L. P. piston. Once the engine moves, steam enters the H.P. cylinder by 27, the proper path, exhausts by 30 and 29, and acting on the large area of the flap 58, opens it, and once more valve 59 is closed by its spring.

Instead of feed pumps, *injectors* are now favoured for feeding locomotive boilers, and two of these, 12, 12, are supplied. They draw from the tender through a strong rubber pipe, and deliver through the clack box 25, in which is a non-return valve. A

PLATE XVIII.



677.

Locomotive.

tern Railway.

GATESHEAD

DESIGNS)

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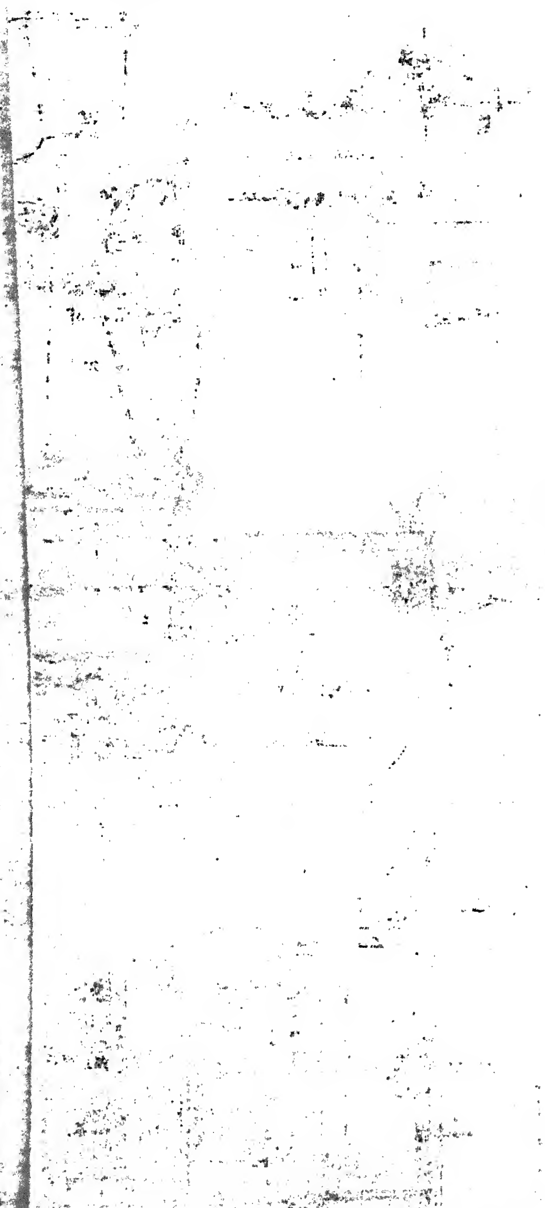


double spring-loaded *safety valve*, 31, is placed over the firebox. The valves are inverted cones, fitting easily, and either centre-point can be lifted by the lever 32 to test the working. A safety link placed within the spring holds the lever in case of breakage. 71 is the steam *whistle*; 70, a lamp bracket; and 72 the chimney, of cast iron. 38 are lubricators for the steam chests.

Tractive Force of a Locomotive is usually taken as the mean pull exerted on the *moving* train, and may be estimated from the principle of work. Thus:

Work given by Steam = Work done on Train.

$$\left.\begin{matrix}\text{Total mean pressure}\\ \text{in both cylinders}\end{matrix}\right\} \times \text{stroke} = \text{Tract. force} \times \left\{\begin{matrix}\text{half wheel}\\ \text{circumference}\end{matrix}\right.$$

$$2 \times \frac{p\,\pi\,d^2}{4} \times l = \mathrm{T} \times \pi r$$

$$\therefore\ \mathrm{T} = \frac{p\,d^2\,l}{2\,r}$$

The tractive force for any particular starting position can only be found by first ascertaining the crank effort for that position (E); then, by moments:

$$\mathrm{T} = \frac{\mathrm{E}\,l}{2\,r}$$

Of course, the greatest value of T must not overcome the adhesive force, or slipping will occur (see p. 571). The tractive force *required* is given at p. 569.

Boiler Fittings.—Boilers having been described at pp. 330 to 339, it remains to consider the principal mountings with which they are fitted. (*See Appendix I., p.* 755; *Appendix II., p.* 833 *and pp.* 899–905; *Appendix III., p.* 918.) (*See also p.* 1148.)

Safety Valves.—Lever-loaded valves, p. 482, are not now in favour, on account of the fear of explosion due to sticking. Directly-loaded valves may be either spring or weight loaded. The former has been shewn at 31, Fig. 675, Plate XVIII.; and a dead-weight valve is given at Fig. 678, as applied to stationary boilers. A casing A, containing the weights, is hung on a cup-shaped valve resting on the conical end of the pipe B. The figure shews also a low-water float C and a high-water float D, which raise rod E whenever the water falls too low or rises too high respectively. Marine valves are spring-loaded, and the

Board of Trade Rule gives half a sq. in. valve area for every sq. ft. of grate surface. (*See p.* 902.)

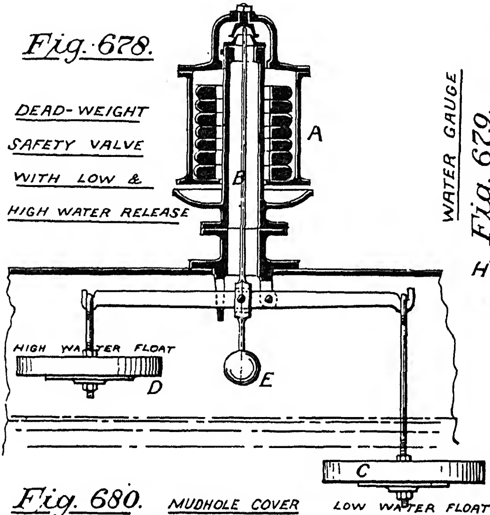
Mudhole Cover.—Manhole covers are merely flat plates covering the raised mountings shewn at Figs. 310 and 311 : mudhole covers, Fig. 680, are more perfect mechanically, the oval plate A being kept closed by the steam pressure, and further secured, by bolts, to bridge pieces B B. The oval shape permits the plate to be entered narrow-ways, after which it is adjusted into position.

Pressure and Vacuum Gauges.—The Bourdon Gauge, Fig. 681, is now generally adopted for both purposes. Within the casing A is a curved tube C, of flattened section, as at D : it is open to pressure at B, but blind at E. If the pressure increase above the atmosphere, the tube distends, and point E moves outward; but a decrease of pressure below atmosphere still further flattens the tube, and point E moves inward. Both movements are transmitted to sector F, which turns, by a pinion, the pointer, thus multiplying the motion; and a hairspring on the pointer axis takes up backlash. The graduations are made by experimental comparison with a mercury gauge.

Injectors.—There are two ways of feeding a boiler with water when under steam: (1) by a pump either driven from the engine, or steam-driven and self-contained, then known as a 'donkey-pump;' (2) an injector may be used. Pumps will be treated in the next chapter. The injector forces water into the boiler without the intervention of moving mechanism, and the only loss is that due to fluid friction, while the delivery of hot instead of cold water is a gain, not to speak of the diminished strain on the boiler. In Fig. 682, A is a section of the instrument, and B shews its application. The injector represented, being non-lifting, must be placed at the tank bottom, but its parts are essentially the same as those of other injectors. C is the water-cock, D the steam cone, E the combining cone, F the overflow pipe, and G the delivery pipe. H is the steam cock, and J a non-return clack box. Cocks H and C being opened, water is forced into the boiler by the steam; and it long remained a surprise to engineers that steam could feed water against its own pressure. The explanation is this: the velocity of efflux of steam

Fig. 678.

DEAD-WEIGHT SAFETY VALVE WITH LOW & HIGH WATER RELEASE



WATER GAUGE

Fig. 679.

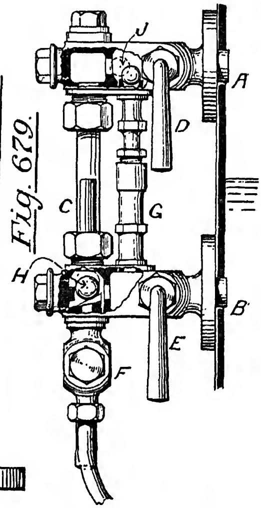
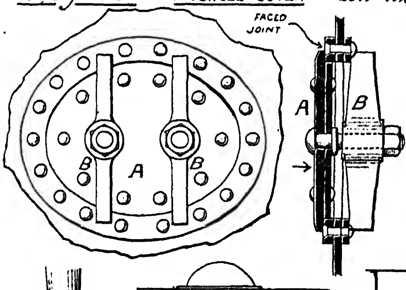


Fig. 680. MUDHOLE COVER



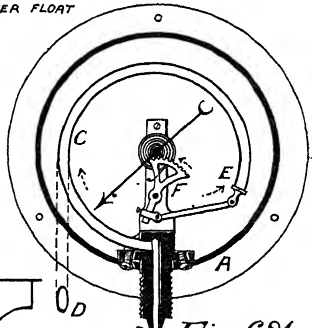


Fig. 681.

BOURDON PRESSURE GAUGE.

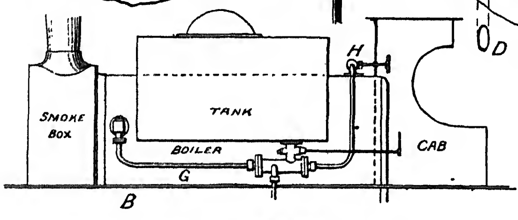
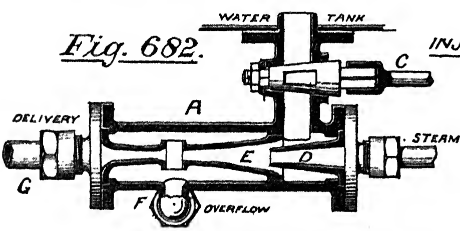


Fig. 682.

INJECTOR



Boiler Fittings.

is some 16 or 18 times that of water at the same pressure, and the jet of steam escaping from D is so suddenly cooled by the tank water through C, that it has not time to reduce its velocity to that due to it as water, and therefore succeeds in piercing the boiler water, carrying the tank water with it. An overflow takes place at F when first starting, which, however, ceases when cocks C and H have been mutually adjusted.

A lifting injector must permit of regulation at the orifice D and ring orifice E, for the conditions of vacuum-forming and water-forcing are quite different, and the former must be first satisfied, after which the latter may be met without disruption of the water column. An ingenious method of automatic lifting injector is now in operation, where the throat E is split longitudinally, and one half hinged near the annulus, the 'flap nozzle' thus formed also causing a re-starting, should the fluid tend to disunite.

Slightly altering the cone proportions, and giving the water a few feet of head, produces an injector workable by exhaust steam; but, on account of the variation in pressure, the flap nozzle must be provided.

Other Mountings for the boiler are: a blow-off cock near the firebox bottom, gauge cocks about 3 ins. above and below the water line, fire bars and bearers, furnace doors, mud plugs, fusible plug in furnace crown to melt in case of overheating, and thus cause the fire to be extinguished, clack box (J, Fig. 682), damper for regulating draught, a filling branch when no other hole is convenient, and sometimes a scum cock. (*See App. II., p.* 902.)

Combustion.—Combustion or burning is rapid chemical combination accompanied by heat and sometimes light. If considerable noise be caused it is termed an explosion. During combination, heat is produced equal to that required to separate the same elements. The separation of carbon and hydrogen, and their re-combination with oxygen, is what the engineer needs to understand, so we will consider the burning of a simple hydrocarbon like marsh gas, shewn by the formula:

$$\text{Marsh gas} + \text{Oxygen} = \text{Carbon dioxide} + \text{Water (steam)}$$
$$CH_4 + 2\,O_2 = CO_2 + 2\,H_2O$$

a case of complete combustion, for no single element remains.

Taking the atomic weights of C, H, and O, as 12, 1, and 16 respectively, we have:

	Marsh gas	+	Oxygen	=	Carbon dioxide	+	Water
	$(12 + 4)$	+	$2\,(16 \times 2)$	=	$\{12 + (16 \times 2)\}$	+	$2\,(2 + 16)$
that is,	16 lbs.	+	64 lbs.	=	44 lbs.	+	36 lbs.
or,	1 lb.	+	4 lbs. gives		2·75 lbs.	+	2·25 lbs.

Again, 1 lb. of carbon burnt to CO_2 gives 14,500 thermal units, and 1 lb. of hydrogen burnt to H_2O gives 62,032 units. In 1 lb. of marsh gas there is $\frac{3}{4}$ lb. of carbon and $\frac{1}{4}$ lb. of hydrogen.

			Units.
∴ $\frac{3}{4}$ lb. Carbon + O	gives	$14{,}500 \times \frac{3}{4}$ =	10,875
$\frac{1}{4}$ lb. Hydrogen + O	gives	$62{,}032 \times \frac{1}{4}$ =	15,508
		Total . .	26,383

Experimentally we obtain a total of 23,582 units, or 2801 units has been required for decomposing the C and H.

Good dry bituminous coal contains on the average, by weight,

Carbon, 83·5 % Hydrogen, 4·6 % Oxygen, 3·15 %

the remaining 8·75 % being Nitrogen and Sulphur, inactive elements. Taking 100 lbs. of fuel the 3·15 lbs. of oxygen is already united to $\frac{1}{8} \times 3{\cdot}15 = {\cdot}4$ lb. of hydrogen as water, and the hydrogen does not assist combustion; so we have left:

83·5 lbs. of Carbon 4·2 lbs. of Hydrogen

Now 12 lbs. of C unite with 32 lbs. of O, or as 1 : 2·66; and 2 lbs. of H require 16 lbs. of O, or as 1 : 8.

	lbs. of O.
∴ 83·5 lbs. C require $83{\cdot}5 \times 2{\cdot}66$ =	222
and 4·2 lbs. H require $4{\cdot}2 \times 8$ =	33·6
Total weight Oxygen for 100 lbs. coal =	255·6 lbs.

or 2·5 lbs. of Oxygen is needed to burn 1 lb. of coal. But air is composed of 77 parts Nitrogen to 23 of Oxygen, by weight.

∴ 23 : 100 :: 2·5 = 10 lbs. of air per lb. of fuel.

Again, we have per lb. of such fuel ·835 lb. of C and ·042 lb. of H,

			Heat units.
∴ ·835 lb. Carbon + O	gives	$14{,}500 \times \cdot 835 =$	12,107
·042 lb. Hydrogen + O	gives	$62{,}032 \times \cdot 042 =$	2,605
		Total units .	14,712

By careful laboratory experiment **one lb.** of such coal is found to have a **calorific value** of 14,701 thermal units, and evaporate 15 lbs. of water at 212°. Also 12 lbs. of air are required per lb. of fuel. (*See pp.* 906 *and* 1148.)

In actual practice considerably less heat is developed, and the evaporation is good at 10 lbs. of water, being commonly 6 or 8. Also 24 lbs. or 312 cub. ft. of air are required, with *natural draught*, to dilute the gases and allow the air to reach the fuel.

Forced Draught.—The essential advantage of forced draught lies in the fact that a smaller dilution of the gases can be allowed, 18 lbs. of air per lb. of fuel, or only $1\frac{1}{2}$ times what the chemist requires. In consequence, a higher temperature is obtained, the grate and heating surface being much more efficient; and thus a smaller boiler will serve the purpose, a great advantage in torpedo boats.

The air must not be solely fed through the fire bars, or a tongue of flame would meet the stoker whenever he opened the fire door. The closed stokehold, the earlier method of solution, places the stoker in a *plenum* of air at a moderate pressure, which enters the furnace as usual. The later method, the closed ash-pit, requires a box-shaped fire door, into which air is fed as well as to the ashpit; but the air to the latter is at a much lower pressure. The air from the box door passes to the coals through holes in the baffle-plate, and the supply is cut off automatically whenever the door is opened. Both methods still have their advocates. The pressure is caused by a fan. (*See App. II., p.* 907.)

Waste of Fuel is largely due to formation of smoke and incomplete combustion, the carbon partly being burnt to CO. Alternate or continuous firing, by careful men or mechanical stokers, and a sufficient supply of air, are the only remedies. The gases also pass up the chimney at a greater heat than 600°,

the best temperature (that of molten lead), for they are also required to cause draught. A quarter of the total heat is thus wasted, and can only be partly obviated by forced or induced draught.

The Gas Engine.—It was early discovered that steam and the steam boiler might be dispensed with if a mixture of gas and air were fired within the cylinder, the arrangement constituting a true heat engine, both pressure and temperature rising at the moment of explosion.

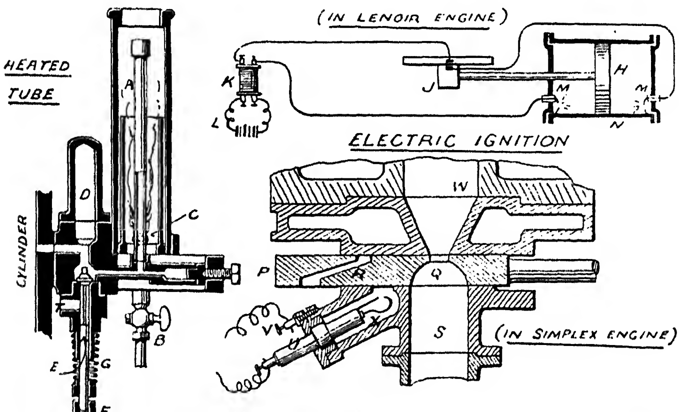
Supposing marsh gas to be mixed with just enough air for complete combustion, a sharp explosion occurs when fired, but if either gas be diluted by too much of the other, combustion is retarded and the explosion is weaker. Dugald Clerk in 1886 made experiments with coal gas and air, and found the sharpness of explosion could be tempered as desired; also, that the pressure was better sustained with slow combustion. This is our experience with heavy ordnance. Treating the gun as an engine cylinder, rapid explosion causes high pressure, followed by pretty rapid fall, and but a small work area is enclosed. Slow burning powder gives a lower pressure curve, which rises somewhat slowly, but is better sustained, and a much larger work area results. Clerk showed that the best working proportions, using lighting gas, was 1 of gas to about 11 or 12 of air.

The first practical gas engine was produced by Lenoir in 1860. It was double acting, charging with air and gas during a half stroke, firing during the remaining half, and expelling the products during the return stroke. Passing over the Otto and Langen engine of 1867, and the Bisschop of 1870, we reach the first commercially successful engine, the Otto "Silent," introduced in 1876 by Dr Otto, who applied the cycle of operations originally proposed by Beau de Rochas in 1862, the strokes being as follows:

1st stroke	outward	→	charge of gas and air
2nd stroke	inward	←	compression of the charge
Dead centre			ignition
3rd stroke	outward	→	expansion of the gases
4th stroke	inward	←	expulsion of burnt gases.

An explosion only occurs, therefore, every fourth stroke, and a heavy fly wheel becomes necessary.

But one detail has caused some trouble to all inventors, the question of igniting the explosive mixture without escape of gas. Three methods have been used:—(1) *Flame ignition,* where a portion of burning gas is carried through an aperture in the slide when the latter is just closing. This method has been used extensively, but occasions frequent misfires when the small aperture becomes carbon coated. (2) *Tube ignition,* Fig. 682 *a.* Here the blind tube A is kept at a white heat by the bunsen flame C, supplied with gas from B, and whenever the timing valve



Modes of Ignition Fig. 682 a.

E is opened by the spring G, the charge, which has been com pressed into the ignition chamber D, then ignites. F is the boss of a lever which keeps valve E on its upper seat, and allows the contents of the tube to be cleared through hole T. Small engines have no timing valve, ignition only occurring when the charge is compressed into the tube. Iron tubes have to be replaced every fortnight at the latest. (3) *Electric ignition* was adopted in the Lenoir engine, but in a faulty manner. The current from battery L was intensified by the coil K. It passed through insulators at M M, and by platinum points through the cylinder N, the circuit being closed by the crosshead J, causing sparks at M M. The covering of the platinum points with carbon or watery vapour

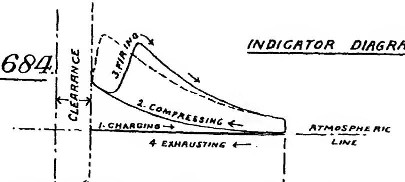
was the cause of failure.* In the Simplex engine a constant shower of sparks takes place in the chamber X, the current passing through the insulator U and back by V. In the figure the cylinder is being charged from S, through Q, but when the slide moves to the right, R connects W with X, and ignition occurs with certainty. (*See App. I., p.* 773, *and App. IV., p.* 963.)

We may now describe the SIMPLEX ENGINE (Système Delamare-Deboutteville et Malandin), Figs. 683 to 688, as a type of a well-designed gas engine. A is the cylinder, supported on the bed plate H, and surrounded by a water jacket B, which also protects the slide casing and exhaust outlet; N is the mixing chamber, and C the piston or plunger. D is the connecting rod, E the crank, F the balance weights, R the crank shaft, and G G the fly wheels, having a pulley P attached for driving purposes. Pipe J is always open to air, and the gaspipe K admits gas when cock L is opened. But such gas is only allowed to enter the cylinder at proper times, viz., when the charging valve M is opened by projection *h* on the slide spindle *g*. As the cycle occupies two revolutions, the shaft Q (which moves the slide *d* backward and forward through the disc crank *f*) makes two rotations to one on the main shaft, and the wheels at R and S together have a velocity ratio of 2 : 1. The charging and ignition having been described the governing and exhaust arrangements remain. Taking the former, shewn in Figs. 687 and 688, the method adopted, as in other gas engines, is to cut out one or more chargings when the engine speed increases. Upon the spindle *h* is a small tapered 'rocker' *j*, and when this is allowed to catch the stem *k* of the charging valve the latter is opened. The governor is a pendulum *n l*, whose lower end is lifted to the right by the rocker *j*, and, being allowed to return freely, its time of fall is invariable. Noting that the rocker *j* is constantly depressed, as in Fig. 688, by a spring, suppose engine speed to be normal, and *j* to be moved to the right, lifting the pendulum. Returning, the pendulum bears slightly upon the rocker, catch *m* lifts *j* to the horizontal, and the valve is opened. But if the slide travel too quickly, *m* misses *j* when returning, and the result is a 'misfire,'

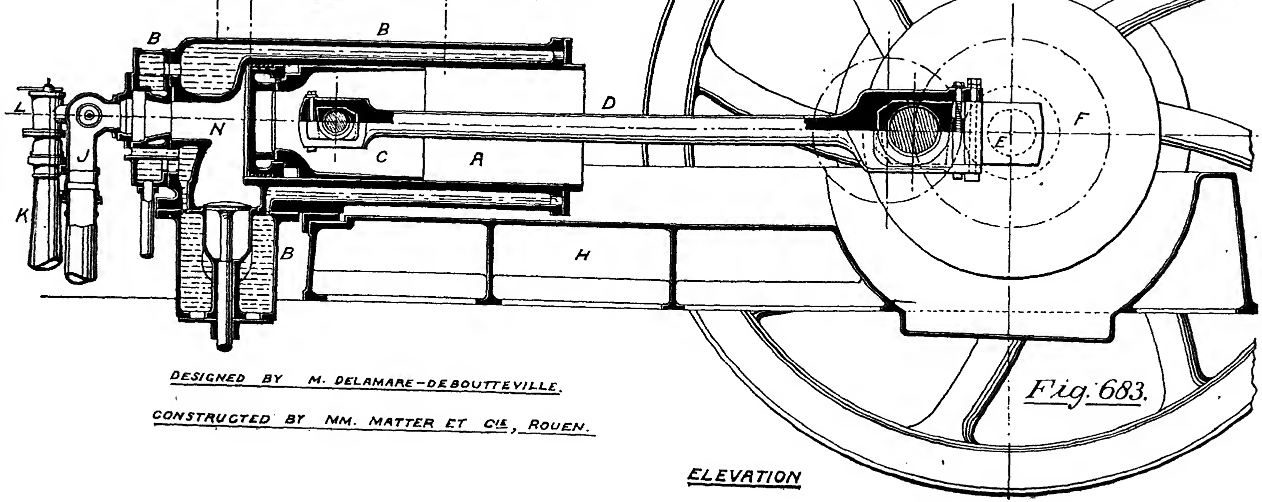
* M. Deboutteville inclines to the former, Prof. Wm. Robinson to the latter cause.

Fig. 684.

INDICATOR DIAGRAM



100. H.P. "Simplex" Gas Engine.



DESIGNED BY M. DELAMARE-DEBOUTTEVILLE.

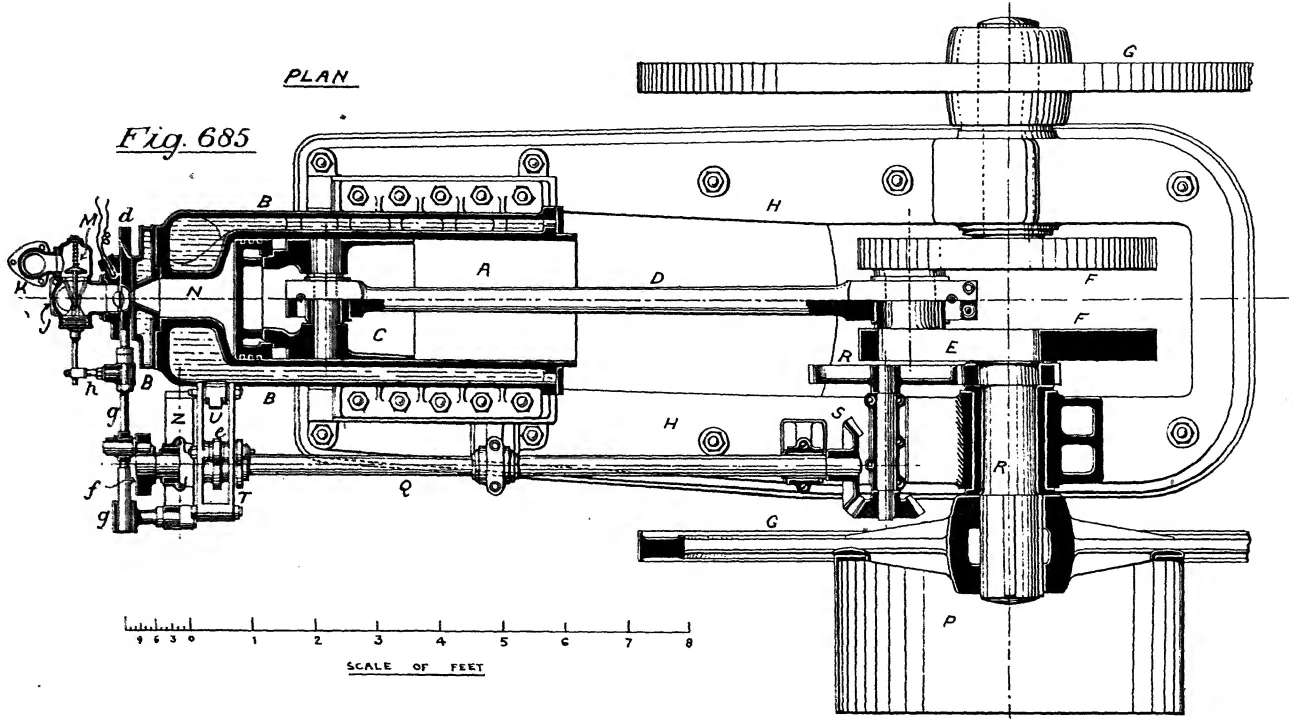
CONSTRUCTED BY MM. MATTER ET CIE, ROUEN.

Fig. 683.

ELEVATION

Fig. 685

PLAN



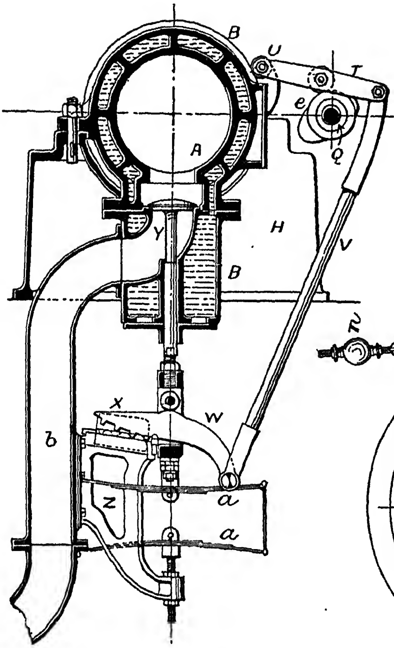


Fig. 686.

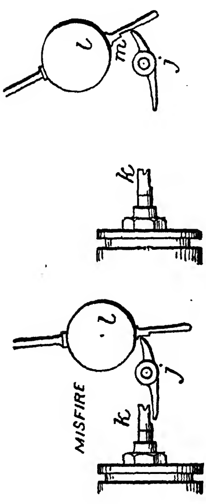


Fig. 688.

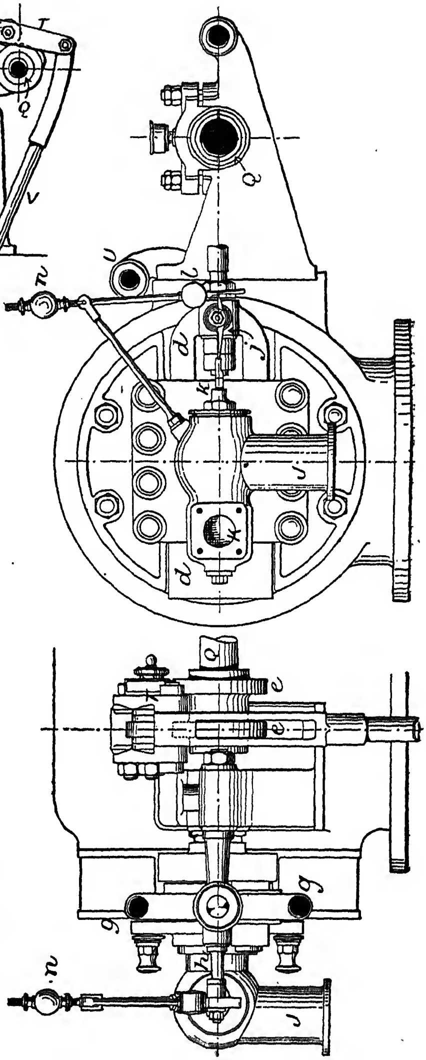


Fig. 687.

Details of Simplex Engine.

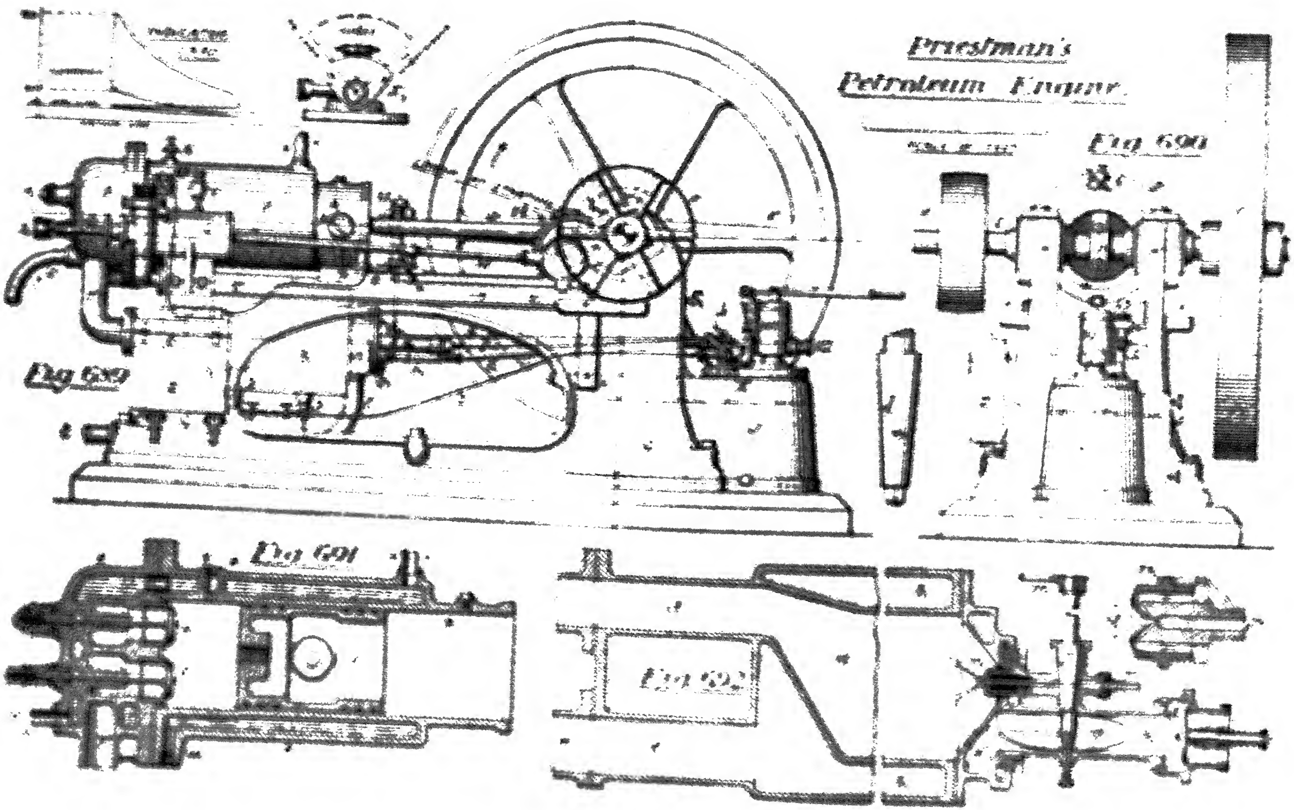
as shewn in Fig. 688. The pendulum may be adjusted to the greatest nicety by raising or lowering ball *n*. The method of opening the exhaust valve is seen in Fig. 686. A cam *e* on the shaft Q lifts the lever T, pivoted at U, and, through rod V, the 'crocodile jaw' W; thus raising the valve against the springs *a a*. W has a shifting fulcrum at X, giving a larger leverage at first, and a quicker opening afterward.

Fig. 684 shews the indicator diagram obtained, which still further illustrates the Otto cycle. One difference in the Simplex working is noticeable; the mixture is over-compressed, that is, a small return motion is made, after leaving the dead point, before ignition occurs, and the force of the explosion only reaches the crank when it is in a better position, viz., at 15° from dead centre.

For the best economy, gas engines should work with 'poor gas,' as produced by the Dowson plant in England, and the Buire-Lencauchez in France: the latter is used in conjunction with the Simplex Engine. Rich lighting gas is expensive for large engines. (*See pp.* 911 *and* 1151.)

Petroleum or Oil Engines, like gas engines, are of the internal combustion type. Petroleum occurs naturally in Russia and America, but is also obtained as paraffin by shale distillation. It is highly complex, consisting of several liquid hydrocarbons having different boiling points: thus, when heated, giving off first the lighter oils, then the burning and lubricating oils, and lastly paraffin wax or vaseline, leaving a residuum. The light oils, including benzoline and naphtha, are dangerous, flashing at or below 73° F.; while the heavy or lighting oils, like kerosene, are thoroughly safe, resisting the flame of a match, or even the electric spark. But the heavy oils are difficult to prepare for the motor, where they are to be intimately mixed with air to form the charge: if vaporised at low temperature, a troublesome residue is formed, while gasification at high temperature produces also tar.

In 1888 Messrs. Priestman Bros. acquired the Etève patents (where *spraying* with air and evaporating in a hot chamber was first proposed), and after considerable experiment produced the first practically successful engine working with safe oil, doing for


Priestman's
Petroleum Engine.
Fig. 689
Fig. 690
Fig. 691
Fig. 692

[illegible] what [illegible] had [illegible]. [illegible] for spraying, a [illegible] was [illegible] the explosive charge, [illegible] being [illegible] with that [illegible] gas engine [illegible]. [illegible] compressed gas [illegible] the [illegible] chamber was kept at [illegible] temperature and [illegible] was [illegible]. [illegible] is adapted [illegible] engine.

Figs. [illegible] show the [illegible] engine, Figs. [illegible] and [illegible] being side and end views respectively. [illegible] the cylinder, [illegible] a jacket, through which water is circulated, entering at [illegible] and discharging at [illegible], and [illegible] an escape cock, removable for insertion of an indicator; [illegible] is the plunger piston, [illegible] the connecting rod, [illegible] the crank, [illegible] the fly wheel, and [illegible] the driving pulley. The main frame [illegible] is cast with the bearings [illegible] and [illegible] caps [illegible], the oil tank [illegible], and the air passage [illegible], from air pump to [illegible] surface. The air pump [illegible], supplying pressure air for forcing forward and spraying the oil, has an adjustable suction valve, and a lubricator [illegible]. The pump rod [illegible] travels at half the speed of the connecting rod, due to gear at [illegible], its prolongation [illegible] opening the exhaust valve [illegible]. A gauge [illegible] shows air pressure, which is regulated by relief valve [illegible]. The igniter [illegible] is an Edison-Lalande primary battery, capable of working four months without attention, the current being intermitted by an [illegible] rod, and while one part of primary wire is connected to terminals [illegible], and passes through porcelain insulators to platinum points within the cylinder, the other part is [illegible] to a brass spring [illegible], and the current closes when the brush [illegible] makes contact with [illegible], firing the charge. No difficulty occurs, as in the former engine, from carbon deposit or water vapour. [illegible] is the spray maker, and [illegible] the vaporizer, the latter kept at constant temperature by the exhaust gases, which escape by pipe [illegible] and chamber [illegible], to the outlet [illegible]. Lamps [illegible] and hand-pump [illegible] are both for starting purposes, while [illegible] and [illegible] are oil or air pipes. Gauge [illegible] shows oil level in tank, and the governor [illegible] acts on the oil admission plug.

Fig. [illegible] is a vertical section of the cylinder, showing [illegible] the inlet valve, and [illegible] the exhaust valve, the former opened automatically by piston suction, and the latter by lever [illegible], [illegible]

[illegible]. The charge is prepared in the vaporiser as shown in *horizontal* section at Fig. [illegible], air [illegible] drawn through inlet pipe [illegible] and valve [illegible] the exhaust passes [illegible] the chamber [illegible]. The [illegible] [illegible] [illegible] consists of [illegible] [illegible] passage [illegible] and [illegible] being [illegible] at [illegible] pressure [illegible] at the [illegible] and are there [illegible] into a vapour or spray filling the chamber [illegible]. The oil pipe [illegible] is seen in elevation at [illegible], and the spraying air enters by pipe [illegible]. The auxiliary air for completing the charge is induced by the suction through valve [illegible] (being there filtered through cotton wool) and along pipe [illegible] to annular chamber [illegible], whence it passes to the vaporiser, and a shutter [illegible] may be adjusted by hand, or closed when standing. Regulation of oil and air is effected by the action on plug [illegible] of the governor [illegible], whose vertical spindle is connected to the lever [illegible], depressing the latter when the balls rise. The oil [illegible] is seen at [illegible] to be pear shaped, closing or opening towards the [illegible] end, never being entirely shut, but full open when the engine is at rest. The throttle valve [illegible] on plug spindle [illegible] [illegible] simultaneously with the oil hole, and thus the air and oil proportions are always correct.

The difficulty of starting is simply overcome. [illegible] pressure from pump [illegible] forces oil and air by two pipes at [illegible] to the lamps [illegible], the six-way cock [illegible] being turned leftward [illegible]. The lamps are then lit, and the vaporiser made somewhat hotter than usual during about 10 minutes. Meaning [illegible] to start, the fly wheel must be turned till the [illegible] closes at [illegible], and the crank takes the dotted position, and the relief valve [illegible] being screwed down, a pressure of about [illegible] lbs. is produced by the hand pump [illegible]. Cock [illegible] is next opened to spray maker, and the sprayed oil enters the vaporiser for [illegible] or [illegible] seconds. Lastly, cock [illegible] being turned on, a quantity of compressed air passes through [illegible] to complete the charge, which, now having a high pressure, opens the inlet valve [illegible] and ignites, the crank rotating till the next impulse, produced automatically.

Oil supply in [illegible] is sufficient for some [illegible] hours' run, but may be easily replenished by pump [illegible] or by gravitation, a suction pipe being coupled to [illegible]. In the latter case an external tank

is placed above J and is connected to it by two pipes (for oil and air). When the pressure in the two tanks becomes equal the oil runs into J by gravitation. Lubrication is effected in the usual manner, at all parts of the engine except the cylinder; the oil condensed within which is ample for the purpose.

Several forms of oil engines are now made by other firms, but none spray the oil. In some, liquid oil is evaporated in a hot chamber, forming vapour and gas, which is mixed with air and fired as usual; and it is said no deposit occurs in ordinary working. In others, perfect oil-gas is produced, and then exploded with air, but the engine must be often cleaned from tarry matter. (*See Appendix II., p.* 915; *also p.* 1165.)

Oil Engines for Motor Cars.—The requirements of motor cars have developed an engine of very small weight using a light benzine oil called 'petrol.' Further information on these important engines will be found in *Appendix II., p.* 915; *Appendix IV., p.* 963; *and Appendix V., p.* 999; *also p.* 1182.

Trials of Boilers, Steam Engines, Gas Engines, and Oil Engines.—*See Appendix III., p.* 937 *et seq.*

Balancing of Engines.—*See Appendix II., p.* 897, *Appendix IV., p.* 967, *and Appendix VI., p.* 1199.

Hot-air Engines.—*See Appendix II., p.* 915.

Steam Turbines.—*See pp.* 895, 966, *and* 1168.

CHAPTER [illegible]

HYDRAULICS AND HYDRAULIC MACHINES

Fluids are defined by their negative property of non-resistance to change of shape, and may be highly compressible, as gases, or very slightly compressible, as liquids. Hydraulics treats of the flow of water in pipes and canals, and with that liquid assumed incompressible we shall only here concern ourselves.*

Head, Pressure, and Velocity Energy. The atmospheric pressure supporting 30 ins. of mercury, the water barometer has a height of 34 ft.; thus a "head," as it is termed, of 34 ft. balances a pressure of 14.7 lbs. per sq. in., and

$$H = \frac{34}{14.7} = 2.31\,p$$

A vertical gauge tube c, Fig. 693, being inserted in a pipe a, water rises in c to a height proportional to the pressure there, *connecting head and pressure*,

$$PA = GHA$$

$$P = GH \qquad \text{and } H = \frac{P}{G}$$

where P = supporting pressure in lbs. per sq. ft., H = height of column, and A its area, and G = weight of a cubic ft. of water. The latter varies from 62.4 at 39° F. to 59.8 at 212° F. for fresh water, but is usually taken at 62½ lbs., and 64 lbs. for sea water. (*See Appendix V., p.* 1006.)

* For remarks regarding compressibility of water, see pp. [illegible]

To connect head and velocity: a water particle of weight w, while at A, Fig. 694, has a potential energy wH, and when fallen to B a kinetic energy of $\frac{wv^2}{2g}$. Neglecting friction and other losses,

$$wH = \frac{wv^2}{2g}$$

$$\text{and } v = \sqrt{2gH} = 8\sqrt{H} \text{ nearly.}$$

When water flows steadily between reservoirs kept at constant level, any portion of water will, neglecting friction and viscosity, be in possession of an unvarying amount of energy, which may be due to head, pressure, velocity, or all three. In Fig. 695, a pressure column A falls short of level C, a portion of the head energy having become kinetic; and the total head $\mathfrak{H}$ consists of H due to unexpended fall, $\frac{P}{G}$ due to pressure, and $\frac{v^2}{2g}$ due to velocity. Multiplying each by w gives the respective energy, and the *energy in one lb. of water*

$$\mathfrak{H} = H + \frac{P}{G} + \frac{v^2}{2g}$$

An interesting experiment, due to Froude, is given in Fig. 696. Two tanks, A and B, have discharge pipes C and D, the former throttled at E, and the latter expanded at F, causing the velocity energy to become respectively greater or less than at the tank mouth, as shewn by pressure columns. Further, the horizontal pressures at E and at F exactly balance, and there is no tendency to move the pipe.

The Jet Pump.—With sufficient throttling, the pressure may be reduced below that of the atmosphere, the principle employed in Prof. Jas. Thomson's jet pump, Fig. 697. Water, under a good head, enters pipe D, and passing through the nozzle at a high velocity, produces a partial vacuum around it. More water entering at A to fill the gap, the combined streams discharge at B, and thus a field may be drained or other work performed.

Discharge of Water from Orifices.—A tank being emptied through an orifice near its bottom, the volume of water

passing per sec. [illegible] area A. [illegible]

Theoretical [illegible] ft. per sec. [illegible] $= Av = 8A$ [illegible]

But [illegible]

$v = c_v 8\sqrt{H}$

Or, suppose the [illegible] head H_f, a *coefficient of resistance* [illegible] and H_v as measured in terms of the [illegible] head H.

Let $H_f = \rho H_v$

Then $H = H_v + H_f = H_v + \rho H_v = (1 + \rho) H_v$

$$\text{And, } v = 8\sqrt{H_v} = 8\sqrt{\frac{H}{1+\rho}} \qquad (2)$$

Equating the two values, (1) and (2):

$$c_v\sqrt{H} = \sqrt{\frac{H}{1+\rho}}$$

$$\text{and } c_v = \sqrt{\frac{1}{1+\rho}} \qquad \rho = \frac{1}{c_v^2} - 1$$

The above losses all occur *within* the vessel and [illegible] a further loss is caused on account of the diminution of jet area [illegible] contraction, at a distance beyond the [illegible] of half the jet diameter. Taking the *coefficient of contraction* c [illegible] let real area $= cA$; then,

Actual discharge in cubic ft. per sec. $= Q_a =$ [illegible]

$$= 8c_v cA\sqrt{H}$$

$$\text{Or, from (2)} \qquad = 8\sqrt{\frac{H}{1+\rho}}\, cA$$

For simplicity one multiplier may be adopted, the *coefficient of discharge* C, equal to $c_v \times c$ (about [illegible]) and then

$$Q_a = 8\, AC\sqrt{H}$$

All the coefficients are determined experimentally, [illegible] measuring the parabolic form of the jet, [illegible]

(Fig. 698), and C by gauging actual discharge. Fig. 698 shews at A a sharp-edged, at B a re-entrant, at C a cylindrical and external, and at E a bell-mouthed orifice. At B the contraction

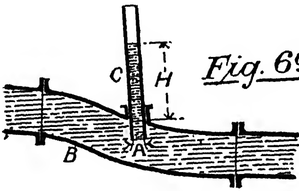


Fig. 693.

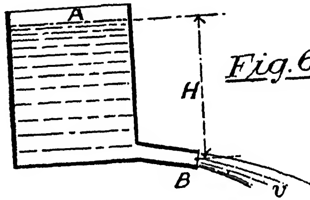


Fig. 694.

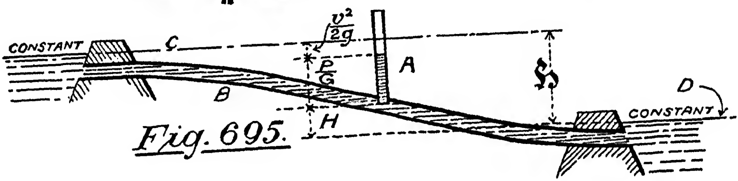


Fig. 695.

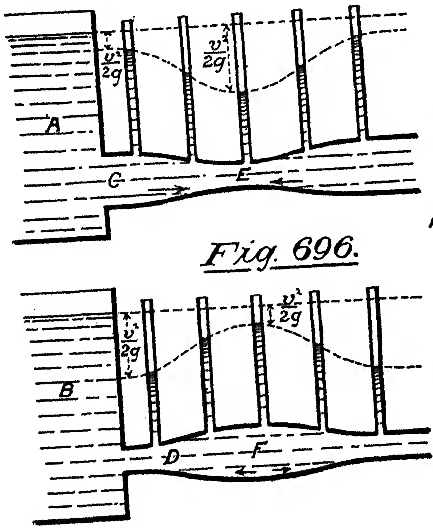


Fig. 696.

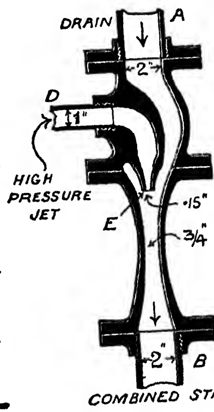


Fig. 697.

Jet Pump

is greatest by reason of the abrupt deviation of the stream lines; at C there is contraction *within* the orifice; and at E no free contraction, so that there C = *c*

TABLE OF COEFFICIENTS (average value).

Orifice is	Sharp-edged.	Re-entrant.	Cylindrical.	Bell-mouthed.
c	·97	1	·82	·99
ρ	·0628	0	·5	·02
κ	·64	·53	1	1
C	·62	·53	·82	·99

Measurement of Stream Horse-Power by Gauge Notches.—Let a stream be partly dammed, the water flowing through the *rectangular notch*, *a b c d*, Fig. 699. To find the discharge, divide H into very small portions h, and treat every small rectangle as a separate orifice, whose area will, when h is infinitely small, be shewn by B. At any depth H_1, $v = 8\sqrt{H_1}$, and discharge through small rectangle $= 8\text{ B}\sqrt{H_1}$. Shewing the various discharges by horizontal lines on base $e f$, the figure is a parabola (the lines $\propto \sqrt{H_1}$), whose base is $8\text{ B}\sqrt{H}$. Then

$$\left.\begin{array}{c}\text{Theoretical}\\\text{Discharge through}\\\text{whole notch, in}\\\text{cub. ft. per sec.}\end{array}\right\} \text{Q} = \begin{array}{c}\text{area of}\\\text{parabola}\end{array} = \tfrac{2}{3}\,8\text{ B}\sqrt{H}\times\text{H} = 5\tfrac{1}{3}\text{ H B}\sqrt{H}$$

$$\text{Actual discharge } Q_a = 5\tfrac{1}{3}\text{ C H B}\sqrt{H}$$

where C, the co-efficient of discharge

$$= \cdot 57 + \{\text{breadth of notch} \div (10 \times \text{breadth of weir})\}$$

Prof. James Thomson adopted the *triangular notch* A, where B/H is constant throughout, suspecting that C would be thereby regular; and he found that $Q \propto H^{\frac{5}{2}}$. Taking an apex angle of 90°,

$$Q_a \text{ per sec.} = 2\cdot 635\sqrt{H^5}$$

where C = ·617 (included in coefficient 2·635). Finally, for *any* notch,

$$\left.\begin{array}{c}\text{Horse Power}\\\text{of Stream}\end{array}\right\} = \frac{\text{foot pounds per sec.} \times 60}{33{,}000} = \frac{\text{Q G}}{550} \times \left\{\begin{array}{c}\text{available}\\\text{height of}\\\text{fall in}\\\text{feet.}\end{array}\right\}$$

The head H is determined by a stake placed in still water above the notch. (*See p.* 1201.)

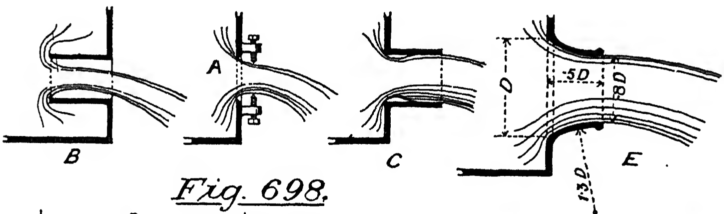


Fig. 698.

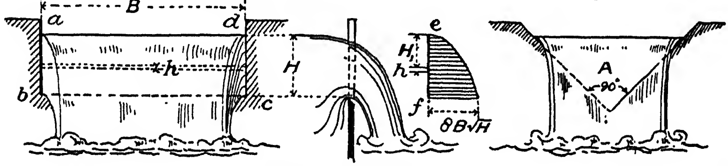


Fig. 699.

Fluid Friction.—The general laws, p. 557, state that $F_n \propto v^2$, and is independent of pressure, but depends directly on the wetted surface. Measuring the surface area A in square feet,

$$F_n = \mu A v^2$$

at moderate speeds, where $\mu = \cdot 004$ for clean varnished surfaces, and ·009 for a medium sand-paper texture (Froude).

Friction in Pipes is principally due to surface or *skin* friction, viscous resistance being extremely slight. Assuming $G = 2g$ approximately, and placing these values so as to cancel,

$$\text{Total } F_n = \mu G A \frac{v^2}{2g}$$

Supposing, now, a piece of water of length L and diameter D of the pipe, is being pushed through the latter at velocity v:

$$\left.\begin{array}{c}F_n \text{ per sq. ft. of} \\ \text{sectional area}\end{array}\right\} = \mu G \frac{\pi D L}{\pi D^2 \div 4} \cdot \frac{v^2}{2g} = G \frac{4\mu L}{D} \cdot \frac{v^2}{2g}$$

As $H = \dfrac{\text{Press. per sq. ft.}}{G}$, we divide by G, and obtain

$$\text{Head lost in friction} = \underline{4\mu \frac{L}{D} \cdot \frac{v^2}{2g}}$$ (*See Appendix IV., p.* 965.)

Experiments on pipes give [illegible] — [illegible] and it is more correct when calculating, to take l a few feet at a time.

Virtual Slope. Water being discharged from reservoir A, Fig. 700, by pipe B C, with a constant velocity through, the [illegible] head may be shown at any place by pressure gauges, [illegible] and [illegible], and in any particular gauge it [illegible] head lost due to friction. This varying head is a straight line [illegible] the water columns, and is called the line of *virtual slope* or *hydraulic gradient*. Suppose the pipe be laid along K K, pressure head would be constant, which is as though the pipe were level, but frictionless; but however it is laid, it is unalterable, only deviating with change in pipe diameter. After crossing the line at c, the pressure within the pipe is less than atmospheric, and the water tends to separate, the tendency becoming a certainty at s.

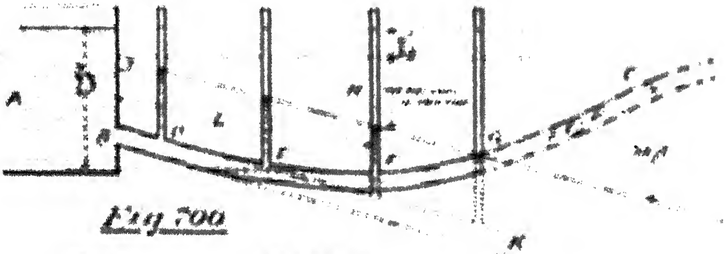


Fig. 700

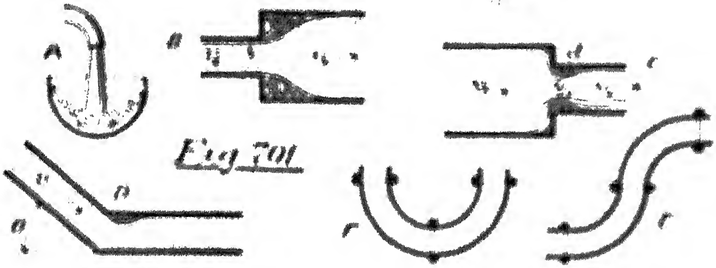


Fig. 701

Loss by Eddies and Shock. Water poured into a basin, as at A, Fig. 701, delivers all its energy as shock; but whenever a sudden change of velocity occurs, eddies are formed which absorb energy. Pipe B, suddenly enlarging, decreases the water velocity,

forming eddies at the corners, and the relative velocity being $v_1 - v_2$,

$$\left.\begin{array}{l}\text{Loss in foot-pounds}\\ \text{per second}\end{array}\right\} = \frac{w(v_1 - v_2)^2}{2g}$$

where w = weight of water passing per second.

$$\text{Loss of head} = \frac{(v_1 - v_2)^2}{2g}$$

At c the water velocity is decreased, but the loss is not as before. There is a very small loss from contraction at d, but the loss by changing the velocity from v_2 to v_1 must be considered.

Any sudden deviation as at b causes loss of energy, probably by eddies. Adopting the formula

$$\text{Loss of head} = \zeta \frac{v^2}{2g}$$

experimental values of the coefficient ζ may be obtained:

θ =	20°	40°	60°	80°	90°	100°	110°	120°	130°	140°
ζ =	.046	.139	.364	.74	.984	1.26	1.556	1.861	2.158	2.431

When possible, bends should continually deviate in the same direction. Thus case a is worse than at c, for in the former there is full loss from both bends, while at c, though there is full loss from the first, there is very little from the second bend. With gradual curvature there is little loss besides skin friction.

Principle of Momentum. At § [illegible] it was stated that force causing momentum was equal to $\frac{w}{g}f$. In another form,

$$Ft = \frac{wv}{g}, \quad \text{i.e.,} \quad \begin{array}{c}\text{Impulse}\\ \text{exerted}\end{array} = \begin{array}{c}\text{Momentum}\\ \text{generated}\end{array}, \quad \text{for } f = \frac{v}{t}$$

If t = one second, and w = *weight of water passing per second*, $\frac{wv}{g}$ = change of momentum, and $F = \frac{wv}{g}$ = change of momentum, a formula we shall now apply to the pressure on wheel vanes.

Case I. To find the pressure due to a water jet on a fixed plate a, Fig. [illegible]. Measuring v in jet direction, $v = 0$ at the plate. Then,

$$\text{Pressure on plate} = \text{change of momentum} = \frac{wv}{g} - \frac{w \times 0}{g}$$

Case II. Let the plate move in the direction of the jet, as at B. Weight of water passing per sec. $= GA(v_1 - v_2)$.

$$\begin{pmatrix}\text{Pressure on} \\ \text{plate}\end{pmatrix} = \begin{pmatrix}\text{Momentum before} \\ \text{impact}\end{pmatrix} - \begin{pmatrix}\text{Momentum after} \\ \text{impact}\end{pmatrix}$$

$$P = \frac{GA(v_1 - v_2)v_1}{g} - \frac{GA(v_1 - v_2)v_2}{g} = \frac{GA(v_1 - v_2)^2}{g}$$

Case III. The reaction wheel is only different from the last case in that the plate pushes the water and the plate pressure is caused by reaction. Ships driven by water reaction, as was the *Waterwitch*, are also similarly calculated, and the best conditions occur when $v_2 = \frac{1}{2} v_1$.

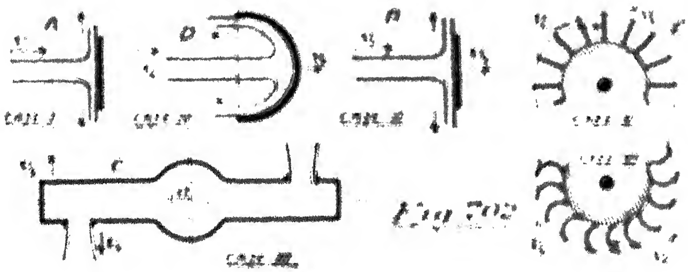


Fig. 702.

Case IV. A moving hemispherical cup. Relative velocity of jet and float when meeting is $v_1 - v_2$ (forwards), and when leaving is $v_1 - v_2$ (backwards), so absolute discharge velocity is cup velocity minus relative backward velocity $= v_2 - (v_1 - v_2)$.

Absolute velocity of jet before impact $= v_1$

" " " after " $= v_2 - (v_1 - v_2) = 2v_2 - v_1$

Weight of water per second $= GA(v_1 - v_2)$

P = difference of momentum

$$= \frac{GA(v_1 - v_2)v_1}{g} - \frac{GA(v_1 - v_2)(2v_2 - v_1)}{g} = 2\frac{G}{g}A(v_1 - v_2)^2$$

If $v_2 = \frac{1}{2} v_1$, absolute velocity of rejection is zero, and all the jet energy is expended on the cups.

Case V. The wheel has a large number of vanes such as B. Then a plane is constantly before the jet, and relative velocity is v_1.

$$\text{Weight of water per sec.} = wAv_1$$

$$\text{Momentum before impact} = \frac{wAv_1 \cdot v_1}{g}$$

$$\text{Momentum after impact} = \frac{wAv_1 \cdot v_2}{g}$$

$$P = \text{difference} = \frac{wAv_1(v_1 - v_2)}{g}$$

giving the general rule: *pressure on normal floats of water wheel = weight of water per sec.* $\times \frac{(v_1 - v_2)}{g}$.

Case VI (c, Fig. 202), is a similar modification of Case IV. Relative velocity before impact $= v_1 - v_2$, and

$$\text{Weight of water per sec.} = wAv_1$$

$$\text{Momentum before impact} = \frac{wAv_1(v_1 - v_2)}{g}$$

$$\text{Momentum after impact} = \frac{wAv_1\{-(v_1 - v_2)\}}{g}$$

$$P = \frac{wAv_1(v_1 - v_2 + v_1 - v_2)}{g} = \frac{2wAv_1(v_1 - v_2)}{g}$$

giving twice the advantage of a flat plate.

Best form of Vane. — a b, Fig. 203, is the float of an undershot water wheel, receiving the impact of a thin stream e.

Fig. 203

Drawing v_1 the velocity of water jet, and v_2 that of the float tangentially, the completion of the parallelogram gives v_r the relative velocity, in magnitude and direction, in which the float

should be made tangential. This form is the same as that of Poncelet, and the action is essentially as at [illegible]

Water Wheels. The earliest forms of water wheels consist of (1) those rotated by water falling down the face, known as *weight machines*; (2) those actuated by water impact on their lower floats, and called *impulse machines*. Overshot and breast wheels belong to the former, and undershot wheels to the latter class.

The Overshot Wheel, Fig. 704, is suitable for falls of 10 ft. to 70 ft., with a discharge of 5 to 25 cub. ft. per sec.; A is the supply, B the tail race, and C the regulating sluice. Fairbairn improved this motor by driving, from teeth upon the rim, a pinion so placed as to receive nearly all the weight of the driving water. Previously all the power had been transmitted through the axle. The efficiency of the machine is about .75. Taking Q as discharge per second,

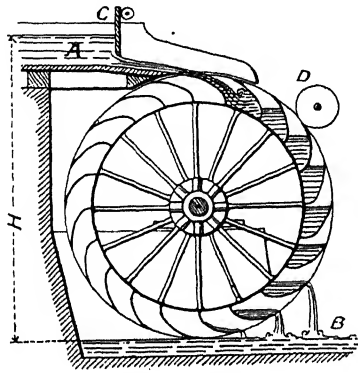
$$\text{H.P.} = \frac{GQH \times 60}{33{,}000},$$

and the water velocity will be slightly greater than that of wheel rim.

The Breast Wheel, Fig. 705, is there shown in its greatly improved state, as due to Fairbairn. The breast B, lying within [illegible] of the wheel, keeps the water in the buckets through a greater length of rim than in the overshot wheel, permitting its escape into the tail race with but little velocity. The regulating sluice is adjusted by a governor, and the penstock C is provided with guide blades to direct the motion of the entering water. The buckets are 'ventilated,' that is, are partly open to the wheel interior, thus permitting air to pass out or in whenever the water enters or leaves respectively.

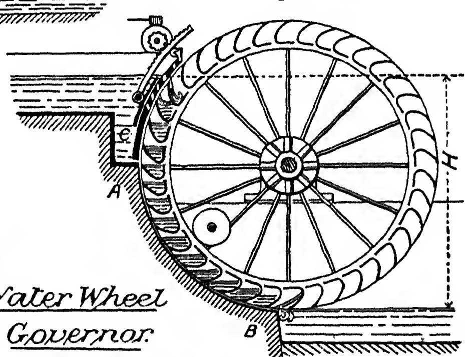
The Governor, Fig. 706, is of the Watt type, but the movements of the sleeve A merely direct instead of actually cause the movements of the sluice. Spindle B, hollow in its lower portion, carries loosely the mitre wheels C and D, each gearing with wheel E on the sluice shaft. When the balls rise, sleeve A lifts by rod H the clutch F, and C being thus put in gear, the shaft G is rotated to close the sluice. If, conversely, speed

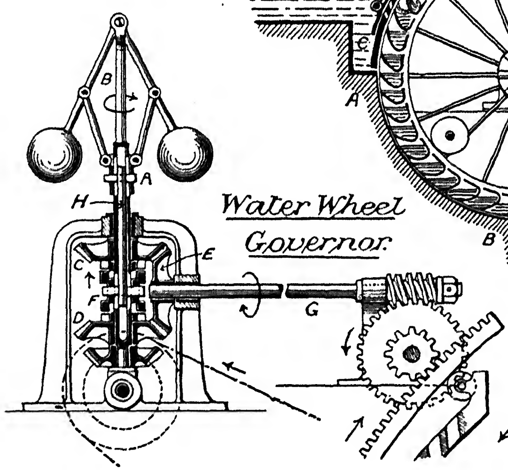
decreases, the balls fall and put D in gear, thus turning G oppositely, and partly opening the sluice. The governor is driven from the water-wheel by a belt.



Overshot Wheel.
Fig. 704.

Breast Wheel
Fig. 705.

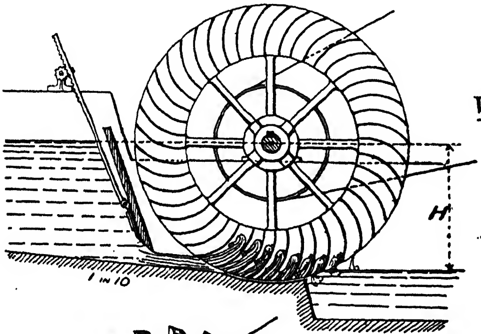




Water Wheel Governor.
Fig. 706.

The Undershot Wheel is shewn in Fig. 707. The form of float has been drawn at Fig. 703, and there only remains to add that, with Poncelet's improvements in floats and race, the water

leaves the wheel with little absolute velocity, and the efficiency is about ·66, a great improvement over that of the old radial-float wheel, which was only ·3. As the water never fills the vanes, there is no pressure, but pure impulse only, and the efficiency is therefore constant under varying sluices. Horse-power may be reckoned from head or velocity (*see pp.* 719 and 720). The circumferential velocity is about ·55 of that due to head, and the jet thickness is about 8 or 10 ins. The wheel is suitable for falls up to 6 feet, and the diameter may be four times the fall.



Undershot Water Wheel

Fig. 707.

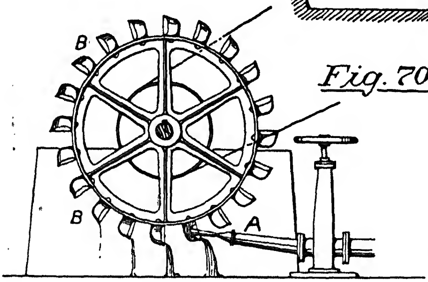
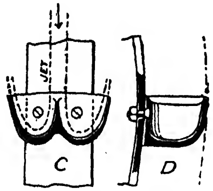


Fig. 708.

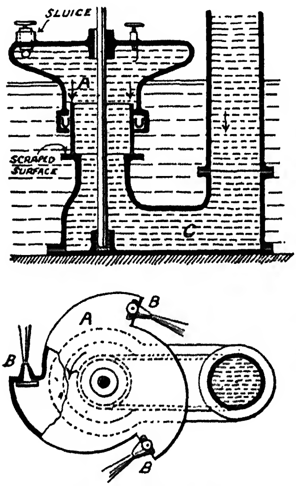


SECTION OF CUP

Pelton Wheel.

The Pelton Wheel, Fig. 708, is an American machine, in which a small jet issues from a nozzle A, with great head, and impinges on a series of cups B B, of the form of a split semicircle in end elevation C, and simply cup-form in side elevation D. In this way the jet, about $\frac{3}{8}''$ diameter, is split, and returned without serious shock. In one example 320 H. P. was given off from a fall of 523 ft., the nozzles being one inch diameter. The efficiency is commonly ·8, but may reach ·9.

Turbines, formerly including only horizontal types, is the term now applied to all water wheels in which a relative movement of the water to the wheel causes reaction. The Reaction wheel, Fig. 709, is the earliest form, being a turbine without guide blades. The casing A, or wheel proper, has tangential nozzles BBB, through which the water leaves, entering at C; its reaction on A thereby producing motion. If the best velocity, that due to head, be employed, an efficiency of ·6 is attainable; but otherwise there is considerable waste of energy. This fact led to the introduction of guide blades and curved vanes, and the invention of the true turbine.



Reaction Wheel,
BARKER'S MILL, OR SCOTCH TURBINE
Fig. 709

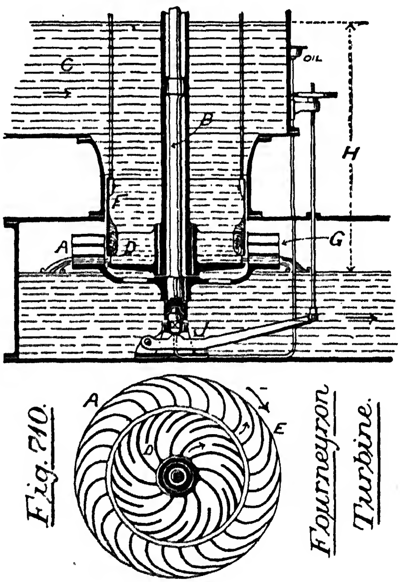


Fig. 710. Fourneyron Turbine.

The Fourneyron Turbine, Fig. 710, is an outward-flow and also a pressure turbine, the wheel passages being kept full. A, the wheel, is keyed to shaft B to transmit the power, and the water flowing downward from C is so deviated by fixed guide blades DD, that it enters the wheel nearly at a tangent. The wheel vanes are so curved that the flow is then changed to a radial direction, the

water leaving with little absolute tangential velocity, having given some 70 or 80 % of its energy to the wheel. Regulation by throttling always reducing the efficiency considerably, the wheel is divided by horizontal plates at G, so that in the drawing there are three separate turbines which can be shut off in succession by lowering the hollow cylinder F. Oil is supplied to the footstep J through a pipe, but immersed footsteps are now superseded. Horse-power may be found either by head or impulse formulæ.

The Jonval Turbine, like the Fourneyron, is a pressure turbine; but while the latter works best above tail water, the Jonval is always drowned or else connected to tail water by a 'suction' tube not more than 30 ft. high, and therefore full of water. Thus a certain head may be saved, which might be lost, through compulsory position of the turbine. Fig. 711 is a vertical section, where A is the wheel, B the guide blades, and C the shaft; and the water flowing parallel to the shaft gives the title 'parallel flow' to this class of turbine. Regulation, formerly effected by throttling, is now preferably obtained by closing a number of guide passages, preserving complete admission for the remainder. In the figure the guide passages form concentric semicircles G G in plan, and are so bent in elevation as to meet the wheel passages A A, which form a complete circle in plan. This arrangement provides retiring room for the sluices F F.

The Girard Turbine was introduced to provide against the loss of efficiency which always occurs when pressure turbines work with fractional supply. This fault being due to the attempted driving with a pressure for which they were not designed, Girard widened his wheel passages towards the outlet, and ventilated them so as never to entirely fill them with water. The energy is then purely due to velocity, and the turbine is an impulse machine; it has also a parallel flow and complete admission to whatever guide passages are open. In Fig. 712, A A are the guide blades and B the wheel. The latter is keyed to the hollow shaft P F, which, continued upward, joins the solid shaft G and transmits the power. The whole is hung on a pivot bearing J carried on the fixed pillar H, and the same arrangement appears in Fig. 711. The guide passages may be closed by vertical shutters K K, whose rods are coupled to rollers

Jonval Turbine

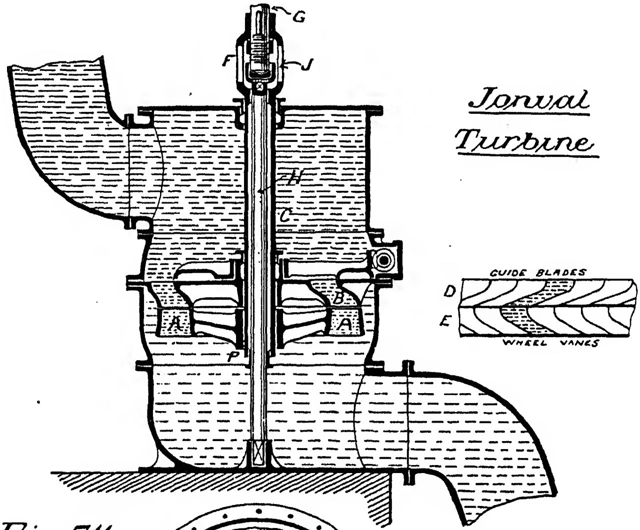
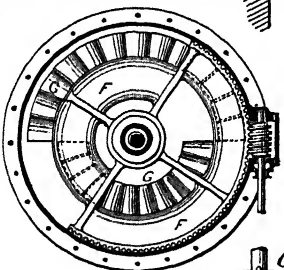
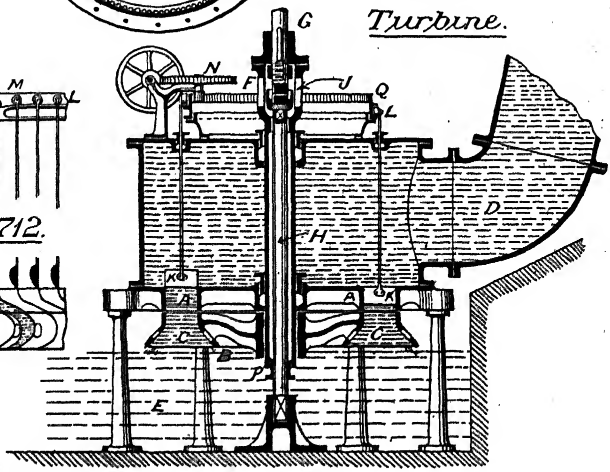


Fig. 711.



Girard Turbine.



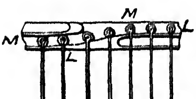
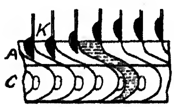


Fig. 712.



L L lying in the groove M M; and as the ring Q is revolved, by hand or governor, through gear N, the shutters are completely raised or lowered, according to direction of rotation.

In Fig. 713 the actual path of the water is shewn in a Jonval turbine at A, and in a Girard turbine at B, *a b* being free path and velocity due to guide blades, and *b c* the wheel velocity; *a c* is the relative velocity, and shews actual path in general direction. Making *c d* = *b c*, *a d* will be the line of wheel vane causing curved water path *a c*, the horizontal ordinates of curvature on *a d* and *a c* being equal.

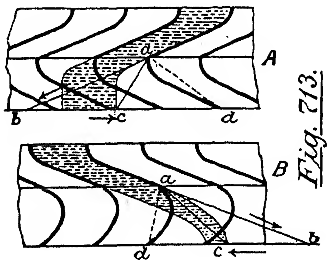


Fig. 713.

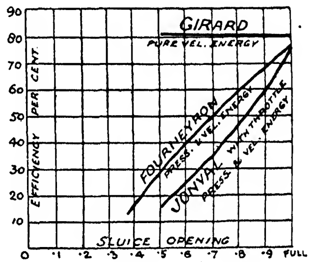


Fig. 714.

Fig. 714 is a diagram shewing comparative efficiencies under varying openings. Although the Girard is usually less efficient than pressure turbines with full sluice, its efficiency is unimpaired by fractional opening.

Thomson's Turbine, Fig. 715.—Here the supply water A enters the rim of the wheel B, and escapes axially into C the tail race, so the machine is called an inward-flow turbine. Its energy is largely due to pressure, the outlet being either drowned or connected with a suction pipe. Referring to the plan, the guide blades D D are pivoted at E E, and can be moved in or out by the levers and links F F. Then the vertical shafts at F F are all connected, and rotated, through worm gear, by the hand wheel G; thus more or less water may be admitted to the wheel. Although the gear is complicated, its action is very perfect, the supply being regulated without materially affecting angle of blades or other conditions, and a nearly maximum efficiency of 75% obtained for all openings. The wheel is shewn in detail at H.

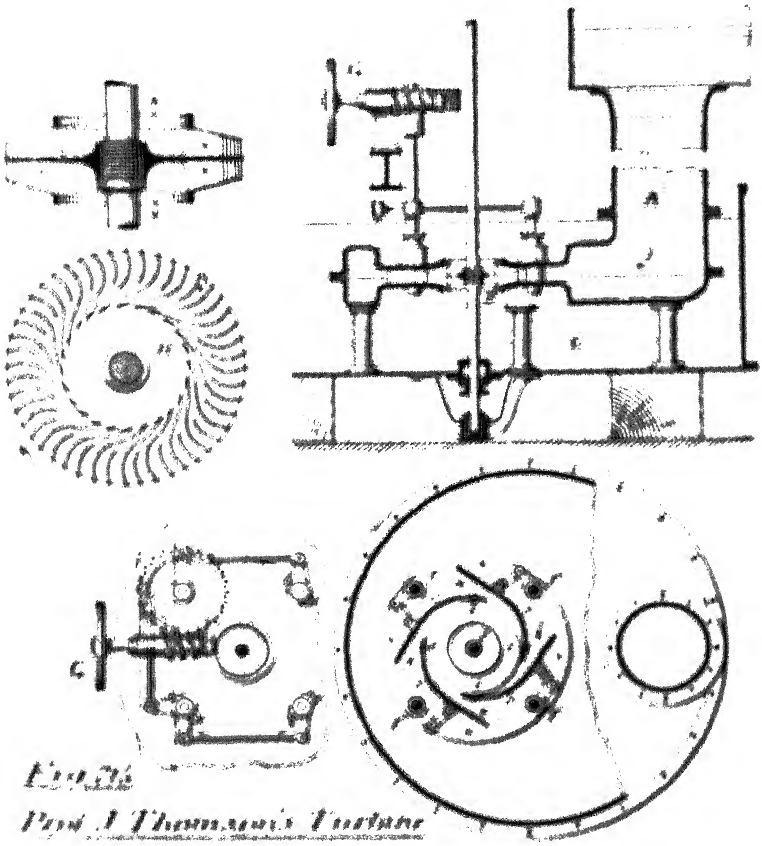


Fig. 26.
Prof. J. Thomson's Turbine

Turbines may be finally classified as follows.

Pressure or Reaction Turbines

Wheel passages filled.
Energy largely due to pressure.
Discharge usually below tail water, or into suction pipe.

Flow
- Parallel or axial (Jonval)
- Outward (Fourneyron)
- Inward (Thomson)
- Mixed, viz. inward and parallel (Schiele)

Impulse Turbines, or Turbines of Free Deviation

Wheel passages never filled.
Energy entirely due to velocity.
Discharge above tail water.
No suction pipe.
Parallel or axial (Girard)

As the water has no forward [illegible] the wheel, each pound suffers a change of [illegible] $\frac{v}{g}$, where v is the forward component of the existing velocity [illegible].

$$\text{Forward pressure on wheel due to each pound of water per sec.} = \frac{v}{g}$$

Multiplying by v_1, the velocity of wheel rim, the work per lb. of water $= \frac{v v_1}{g}$ foot-pounds per second. Now the energy given per pound of water is H foot-pounds, of which ηH is given to the wheel:

$$\eta H = \frac{v v_1}{g}$$

the fundamental equation for turbines. The object also of curving the guide blades is to give a large forward velocity to the water, and the wheel blades are so curved as to reduce that velocity to zero, thus giving all possible energy to the wheel.

The Centrifugal Pump is simply a reversed turbine, for while water enters the wheel eye without whirl, it receives a considerable tangential velocity by the time it reaches the wheel rim. The pressure thus given to the water being found from momentum, increase of head is obtained; then, from the previous paragraph, taking v = tangential component of water velocity when leaving, and v_1 the peripheral velocity of wheel,

$$\frac{v v_1}{g}\eta = H$$

where η is about [illegible]. Fig. [illegible] gives the sections of a pump to lift [illegible] ft., with a pipe velocity of [illegible] ft. per sec. The water path from eye to circumference is that of a free vortex, the form being a logarithmic spiral, and the direction of wheel rotation is shown by the arrow. The casing B is a volute-shaped pipe of increasing diameter towards outlet, to accommodate itself to quantity of water passing; D D are hand holes, and cover E is removable so that the wheel may be withdrawn when required. F is a small 'whirlpool chamber' to let the change from velocity to pressure energy be made with little shock. In starting, sluice [illegible] is closed and air exhausted by a steam jet through ejector G, when water

rises; the sluice is then slowly opened while the shaft is being rotated, and pumping becomes continuous (scale of figure $\frac{1}{42}$).

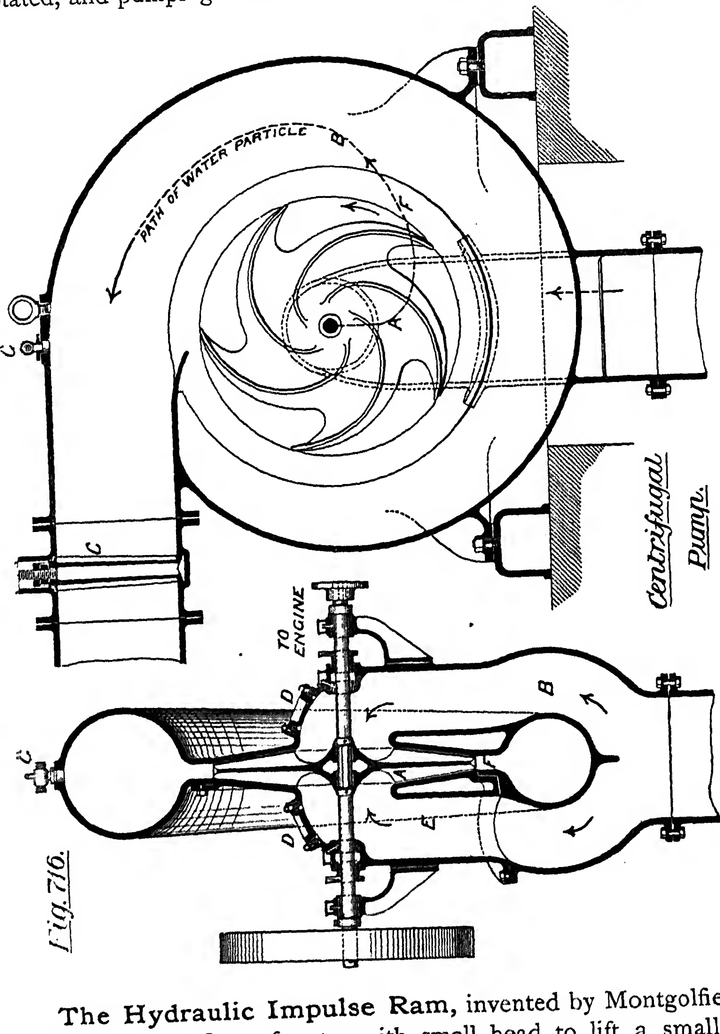


Fig. 716. Centrifugal Pump.

The Hydraulic Impulse Ram, invented by Montgolfier, enables a large flow of water with small head to lift a smaller quantity against greater head; and is commonly used to provide

water supply from a low stream to a [illegible] valve A is slightly heavier than the static pressure of the [illegible] water, which enables the latter to pass [illegible] velocity. This [illegible] and the [illegible] which opens, allowing a portion of the [illegible] to pass [illegible] water at [illegible] again endeavouring [illegible] action is repeated. The air vessel [illegible] steady [illegible]

Piston Pumps. The *Suction or Lifting Pump*, Fig. [illegible] is placed at not more than [illegible] ft above supply water. Piston [illegible] carries a delivery valve, and [illegible] is a suction or foot valve, both opening upward. Then the machine is at first an air pump

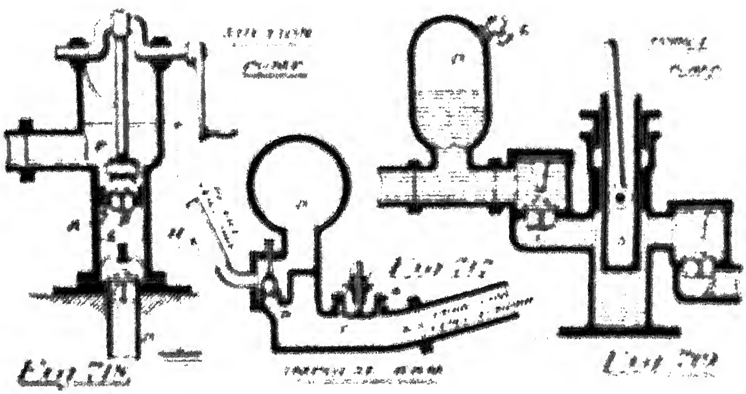


Fig. 718. Fig. 717. Fig. 719.

producing a partial vacuum within [illegible], which at once fills with water from [illegible]. When high enough, water is simply lifted from B to F, and flows away. The resistance to upward motion is due to a column of water of height H, and base equal to piston area.

The *Force Pump*, Fig. 719, is adapted for greater heads, only a small portion of the work depending on vacuum formation. When plunger A is raised, water enters through the suction valve B, and on A's descent this water is forced through the delivery valve C, the head depending on pressure exerted. Air in the vessel D is compressed during delivery, and acts as a spring to continue the flow, while the plunger is occupied on the suction stroke. Cock E is used to re-charge with air when the latter is absorbed by the water. A *double-acting force pump* has one delivery and

one suction valve to each end of the cylinder,* and the plunger becomes a piston.

All the preceding may be driven by steam power. The oldest steam pump yet in use is the Cornish engine A, Fig. 726. Its pumps are of the lift type, arranged in relays with less than 30 ft. between each pair, and the water is lifted from tank to tank till it reaches the surface. Two forms of 'donkey pump' are also shewn at 4, Fig. 447, and 1, Fig. 448, pp. 486 and 488, where the engine valves are operated from a crank shaft. There is, however, another class of pump which dispenses with the crank, being therefore called 'direct-acting,' and probably the best of this class are those that necessarily work in pairs, being termed 'duplex.'

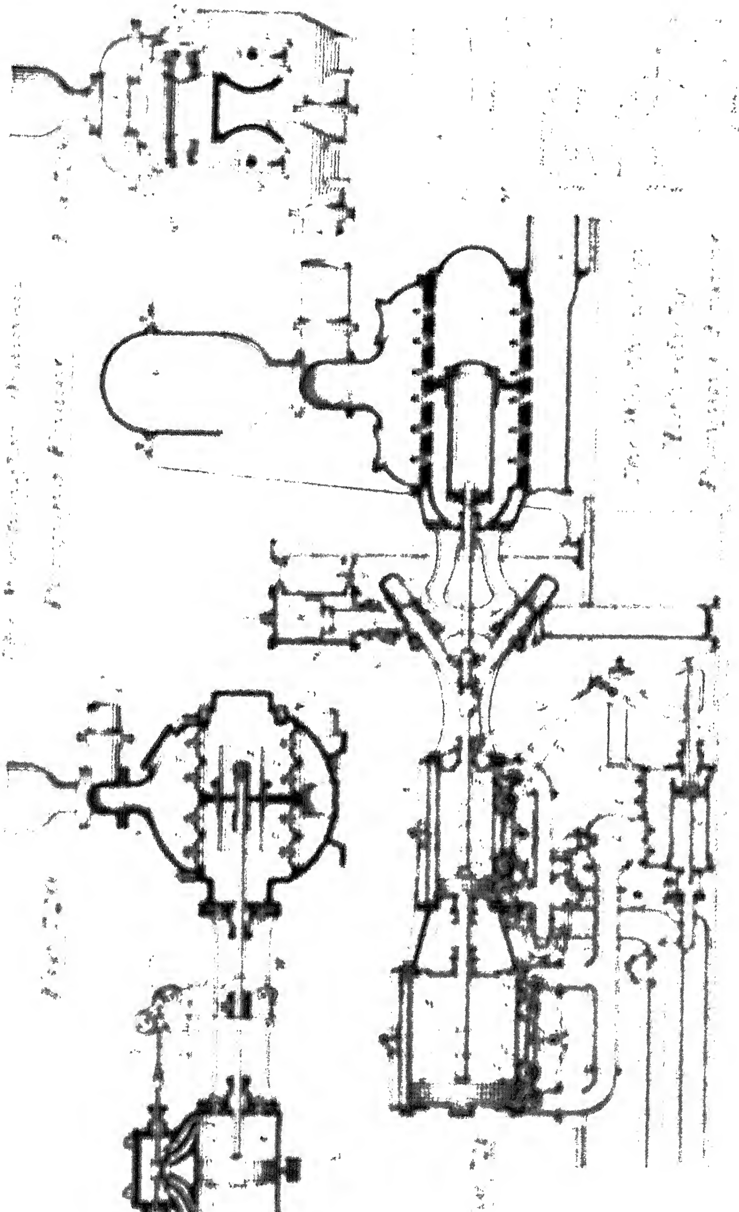
The *Worthington Pump* is a duplex steam pump, its ordinary form being shewn in Figs. 720 and 720*a*. Two steam cylinders side by side at A, have pistons connected directly to two pump plungers at B. When a D valve is employed for an engine working without expansion, the valve and piston strokes cross mutually at half phase, and the piston cannot then directly actuate the valve. In this pump, piston No. 1 works valve No. 2, and piston No. 2 moves valve No. 1, by lever gear, the motion of the two pistons being alternate; thus, levers L and M rock valve-levers *l* and *m* respectively. The valves and pistons are, however, so interdependent, that immediately steam enters either cylinder, the action of the engine commences as a whole, and will continue unless special friction difficulties intervene. To enable each piston stroke to be completed before the valve reopens to steam, the exhaust ports C C are separate from the steam ports D D; a quantity of steam is thus also imprisoned as a cushion. In the pump, E E_1 are the suction, and F F_1 the respective delivery valves, small and numerous, to give sufficient area while diminishing the closing blow. The arrangement, also, enables the pump to both draw and deliver at every stroke, and the contrivance is double-acting; in addition, the air vessel J equalises the flow, and the water leaves at K.

The expansive use of steam has been provided for in the Worthington *high-duty* engine, Fig. 721. The engines are a pair of tandem-compounds, where A is the high-pressure, and B the

* Except in the case of the accumulator pump, Fig. 722.

low-pressure cylinder; and each engine works its neighbour's valves. Thus the lever C of the opposite engine moves the rods D and T, from which a system of link work connects to Corliss valves, the fulcra above and below the cylinders being used respectively for the exhaust valves F F, and steam valves E E. After use in both cylinders, the steam exhausts into the condenser G, from which the water and vapour is withdrawn by the air pump H, and delivered into the hot-well S. The pump itself needs no description, but special attention must be drawn to the means by which the driving force is so equalised as to be nearly uniform when delivered to the plungers. Compensating cylinders L L, or 'pots,' rocking on pipe trunnions, contain water under a steady pressure of about 200 lbs. per sq. in., and have plungers pivoted to the pump rod. This pressure constitutes a resistance to the steam pressure during the first part, and an assistance during the second part of the stroke, much in the same manner as the inertia of the reciprocating parts, and the effect on the work diagram is shewn at M. *a* and *b* are the indicator cards; *c* and *d* shew the pressure exerted by the pot plungers, *c* assisting, and *d* opposing the steam pressure: *e* is the combined effective-pressure diagrams from both cylinders; and *f* is the resultant pressure on the pump-rod after adding *c* and deducting *d*. The pot pressure is kept sensibly constant by the intensifier N, whose larger piston P is under an air pressure of about 75 lbs. per sq. in. from the air vessel K, due to the water column; and the smaller area Q is exerted on the water in the pots. The arrangement constitutes a sort of governor, which controls the pump stroke, shortening it if a pipe happens to burst. To accurately adjust the pot pressure, some air is admitted under P by cock R, causing a pressure of about 35 lbs. per sq. in. These pumps are constructed by Messrs. Jas. Simpson & Co.

The Accumulator Pump, Fig. 722, is a double-acting pump, requiring but one suction and one delivery valve. On account of the great pressure to be resisted (750 lbs. per sq. in.), an air vessel is inadmissible. Referring to Fig. 663, in addition to Fig. 722, the piston A has twice the sectional area of rod B; so when A moves rightward, displacing the whole cylinder volume through delivery valve D, half returns into B, and *half goes to delivery pipe b*. A, returning leftward, draws a whole volume through suction



valve C, none passing D, while the volume in B, or *half, goes to delivery pipe:* thus there is constant delivery, though suction only occurs on alternate strokes. An additional non-return valve E permits each pump to be worked separately. (*See also pp.* 1204 *to* 1210.)

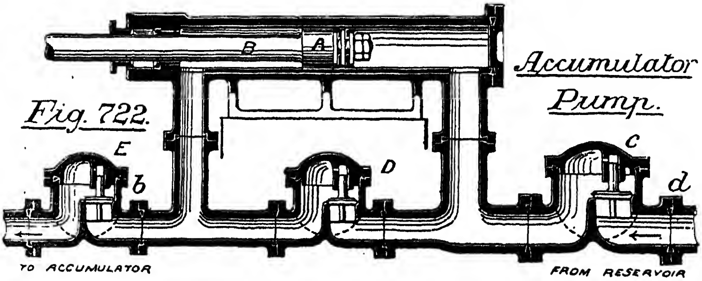
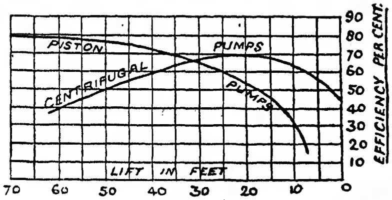
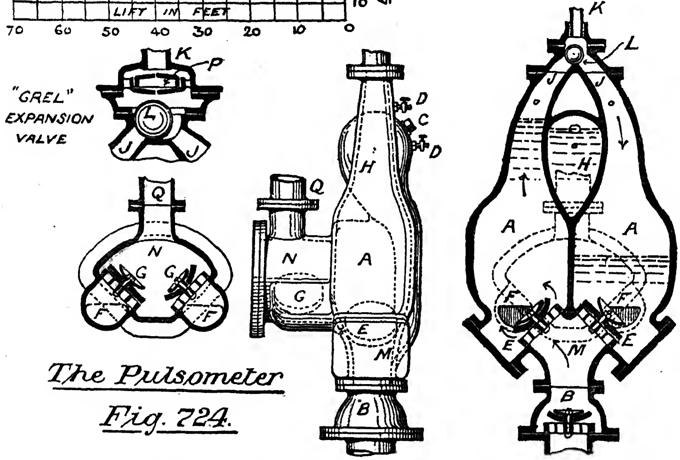


Fig. 722. Accumulator Pump.



Comparison of Pump Efficiencies. Fig. 723.



The Pulsometer Fig. 724.

Pump Efficiencies. — At Fig. 723 is a diagrammatic statement of the efficiencies of centrifugal and piston pumps under different heads, shewing that the former are least efficient

under large head, and the latter under low head. In consequence, simple centrifugal pumps are only employed for pumping large volumes of water under small head, while positive pumps are more suitable for pumping small volumes under great pressure.

The Pulsometer is a pump in which steam acts directly on the water without the intervention of a piston. It is naturally wasteful in working, but is simple and quickly applied on emergency. Referring to Fig. 724, there are two side chambers A A to receive the water alternately, and an intermediate vessel H, whose purpose will be explained. EE are suction and GG delivery valves, B a foot valve, N the delivery chamber, connected to A by short pipes FF, and Q the rising main or delivery pipe. To start the pump, the three vessels are filled through the hole C, the water resting on foot valve B. The ball L being compelled to lie on one or the other seat at J J, steam is admitted at K, and, entering, say, the right-hand passage, displaces the water through F, *without agitation*, until the level falls to the upper edge of the orifice. Steam then blows through into F with some violence, and an *instantaneous condensation* occurs, causing a partial vacuum in A. The ball being now drawn to the right-hand seat, water rises into the right chamber ready for the next stroke, steam enters the left chamber, and the action is continuously repeated. The vessel H, though practically uncharged with air, serves the purpose of an air-vessel, assisting the steady flow into N by the small head of water which it provides; and to prevent the sudden shock caused by the rush of suction water, air-cocks D D are placed on the three vessels, and kept open to a very small amount. The 'Grel' valve at P is often applied to economise the steam supply. It is simply a short hollow piston, which rises and falls on account of the difference of pressure within and without it, thus closing pipe K after a portion of the stroke has been completed. (*See p.* 966.)

The Hydraulic Press may be looked on as the seventh simple machine (see p. 480), and is the basis of the transmissive principle. Fig. 725 represents the press, with pump attached, as used to compress cotton bales. The pump A draws water from the tank B, and forces it, under pressure, to the ram cylinder C, a rapid exhaust being obtained through the relief valve E when required. Let D = diameter of ram, and *d* that of the pump,

while the pump leverage is L : 1 ; from the principle of equal transmission, one pound per sq. inch on the pump plunger is one pound per sq. inch on the ram, and

Total Mechanical Advantage

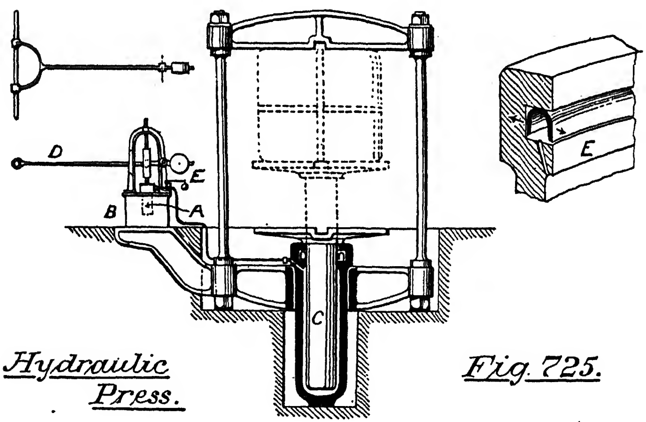
$$= \text{Mech. Adv. of press and pump} \times \text{leverage}$$

$$= \frac{\text{area of press}}{\text{area of pump}} \times \frac{L}{1} = \frac{D^2 L}{d^2}$$

Neglecting friction. Taking pump efficiency $= \cdot 76$, and press efficiency $= \cdot 95$; both combined $= \cdot 76 \times \cdot 95 = \cdot 72$

$$\therefore W = P \times \frac{D^2 L}{d^2} \eta = \frac{\cdot 72 \, D^2 L}{d^2}$$

(*See p.* 1210.)

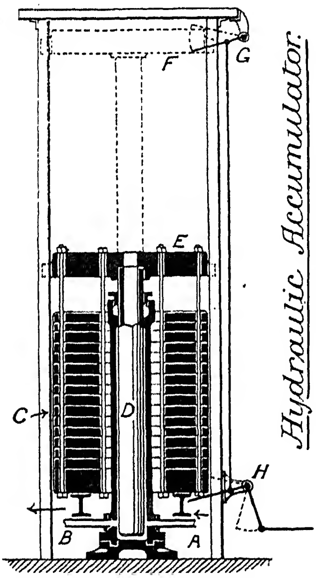


Hydraulic Press. Fig. 725.

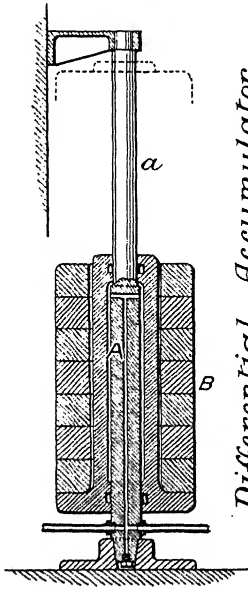
The ram cylinder should be approximately hemispherical (see p. 68), and its strength is found at p. 399. The leather collar E is a most efficient packing, being distended by the pressure water and pressed against the ram surface. The hydraulic jack, p. 206, is simply a miniature press, where G is the ram and D the plunger. Its efficiency is, of course, much higher than that of other jacks.

The Hydraulic Accumulator is probably the most important adjunct in hydraulic transmission, constituting an artificial head, in which the water pressure is caused by other material than water. In Fig. 726, a series of weights at C hang from the

T-head E, and, through ram D, exert pressure on the water within A B. The weights being raised to position F, are a store of potential energy, which may be given out at will through the pipe B. Water is pumped in at A to raise the ram, by an engine such as that in Figs. 663–5, *p.* 682, and the latter is automatically stopped and started from the accumulator, as required, by the levers at G and H, struck by the load. The pressure water



Hydraulic Accumulator. *Fig. 726.*



Differential Accumulator (TWEDDELL'S) *Fig. 727.*

drawn at B may now be applied to the driving of machines doing intermittent work, such as

1. Cranes upon dock wharves, &c.
2. Boiler-shop and shipyard tools.
3. Lifts for hotels, &c.
4. Swing and other movable bridges.
5. Manipulation of heavy guns.

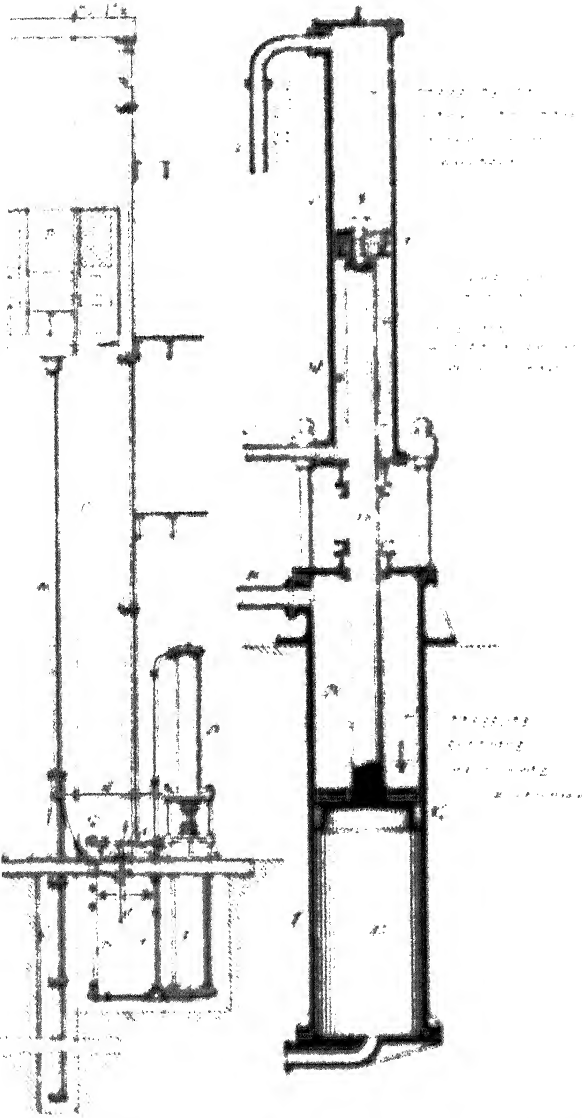
In all these cases the pumping-engine will have sufficient time between shifts to catch up on the machines, and thus a com-

paratively small engine, working all the time, may serve for very heavy work occupying only a short period (see Case 4 especially). It is in the great storing capacity, and the little loss (skin friction being independent of pressure, and water incompressible) that hydraulic transmission is of such immense advantage. The usual large pressure, 750 lbs. per sq. in., is adopted because the friction is then much less in proportion to power transmitted, area of pipe being small. Chapter VII. illustrates hydraulic transmission applied to Case 2, and the student may now refer to pp. 292–3, 301–2, 314, 317, 320, and to Plates XV. and XVI., also to Case 12, p. 580.

Fig. 727 shews Mr. Tweddell's *Differential Accumulator*, where great pressure is obtained by considerably decreasing the ram area. B is the load, and the effective area of ram is A minus *a*. Comparing with Fig. 726, it must be understood that, weights being equal, we lose in time what we gain in pressure, and thus this apparatus is specially suitable for small machines, such as portable riveters. The work stored in any accumulator is *the weight or load, in lbs.* × *the height lifted, in feet*, or

$$wH \quad \text{foot pounds.}$$

A Hydraulic Lift, as devised by Mr. Ellington, and known as a 'balanced' lift, is shewn in Fig. 728. A long ram A, working in a cylinder C, thereby lifts a cage B, and the load consists of (1) the cage, (2) the people or goods, and (3) the ram weight, the last two being variable. In the older and dangerous method the average load was balanced by a weight hung from a cord carried over a pulley, and connected to the top of the cage; but here the cage and people are lifted by separate water columns, while the varying ram weight is supported by a head which similarly varies. The variation in ram weight is due to the ram's varying immersion, the upward support from the water (apart from artificial pressure) being equal to the weight of fluid displaced. Referring to Fig. 728, the pressure from the main is led to the cylinders D and E. Upon piston F is a constant pressure, through L, supporting *weight of cage + ram when down ;* and on piston G, through K, pressure water is admitted when required, supporting the *people + friction*, viz., the nett load. Both these pressures are used to intensify the water in M, which



...ton's Balanced Hydraulic Lift

Fig. 728

is directly connected to the ram, and on account of such intensification the ram diameter can be as small as we please, merely strong enough to prevent its bending. The volume in M being just sufficient to fill the ram cylinder during full stroke, the pistons F and G fall to the bottom of their cylinders, due not only to pressure from main, but to a constantly increasing *weight* of water. It is this weight, due to water, filling nearly the cylinders D and E, which, bearing on pistons F and G, so intensifies the pressure in M, as to support the whole unimmersed ram weight; being clearly a maximum when the cage is fully raised, and nothing when the cage is lowered to the bottom. The varying ram weight is, therefore, correctly balanced in all positions, and the only load to be averaged is that of the people. When lowering, water is exhausted from N, and the descent caused by the weight of the people.

The cord P, passing round a pulley on the working valve Q, will open the latter to pressure *p*, or exhaust *e*, in any position of cage. If the water in M decrease through leakage, the cage is lowered to the bottom, and water at N exhausted: then pressure water being admitted at R, the pistons are forced upward, compelling some water to pass from above to below piston F, through its packing; at other times R is empty. (*See Appendix V., p.* 1006.)

Intensifiers, or intensifying accumulators, are a means of transforming small pressure, as from a town main, into a really useful hydraulic pressure. Recent descriptions will explain the principle, and good examples will be found at pp. 375 and 836.

Hydraulic Cranes have many advantages over others. Being worked intermittently, a small pumping engine will store the power: the latter, again, being used with considerable rapidity and saving of time, a consideration when loading vessels at wharves. The lifting, too, being done without vibration or noise, makes these cranes of special use in raising foundry boxes and other like work. The cranes are also very simple.

Fig. 729 shews a cylinder, ram, and pulleys, the essential apparatus for each motion of a hydraulic crane. Cylinder A has a common stuffing box C, packed with hemp, and carries a number of 'fixed' pullies, $D_1 D_2 D_3$, the ram P supporting an equal number of 'movable' pullies $E_1 E_2 E_3$. To prevent the ram turning on its axis, the head F slides on guides G G, and the

whole apparatus is fixed to the crane by feet J J. A wire rope or chain being attached to the eyebolt K, and carried round the pullies $E_1 D_1 E_2 D_2 E_3$, leaves D_3, by W, to the load or slewing wheel, as desired. Examining by the pulley principle Fig. 439, p. 483, the mechanical advantage will be inversely as the number of cords or chains at L L, Fig. 729, P being now the *greater*, and W the *lesser* force. Neglecting friction,

$$\text{Mech. Adv.} = \frac{W}{P} = \frac{1}{\text{no. of cords}}$$

And allowing for all resistances,

$$W = \frac{P}{\text{no. of cords}} \times \eta$$

where efficiency η varies with the number of pullies, by the following table, found from practice :

VALUES OF η FOR HYDRAULIC CRANES.

NO. OF PULLIES.	0	2	4	6	8	10	12	14	16
η =	·87	·8	·76	·72	·67	·63	·59	·54	·5

and the greater tension, at tail end, equals $P \div (\text{no. of cords} \times \eta)$.

Thus, a heavy pressure with slow speed has lifted a smaller load at greater speed, the distance between pulley centres having been *increased*.

In order that the ram shall finish its stroke quietly, automatic cut-off gear is supplied. Valve H being opened to pressure by raising rod N fully, the ram, ascending, strikes a tappet R by means of the projection Q, when the stroke is nearly complete, thus causing lever M to be pulled over to position S, and closing the valve. A further movement of M to position T opens H to exhaust, and the ram descends by the pull of the load.

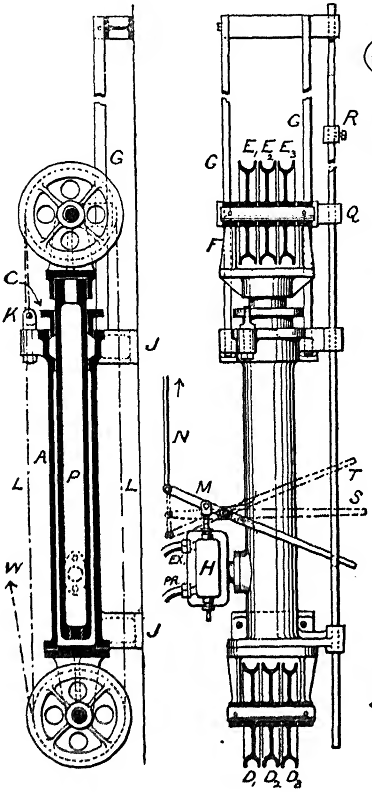
Reference may now be made to Plates XV. and XVI., shewing various hydraulic cranes. That on the left in Plate XV. is the best example of pulley gear. Thus, cylinder D is for lifting, E for traversing, and C for slewing, all worked from valves at S.

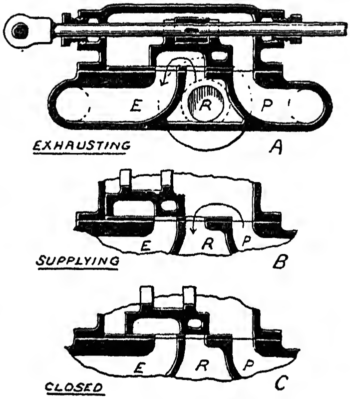
Working Valve.—When a D slide is used, Fig. 730 is the usual form, where P, R, and E are the passages from pressure, to ram, and to exhaust, respectively. At B the valve is open to pressure, and at A to exhaust, while at C the ram passage is entirely cut off, by hand or automatic gear.

Hydraulic-pressure Engines, though wasteful with small pressures and high speeds, may reasonably be used when supplied with water at 750 lbs. pressure or more, the piston speed being not more than 80 ft. per minute. The first piston engine, invented by the late Lord (then Mr.) Armstrong in 1838, was of the rotary type. Subsequently he adopted side-by-side cylinders with reciprocating pistons, and in the present engine, as applied to heavy work, such as turning ships' turrets or swing bridges, there are three oscillating cylinders, whose pistons connect to the same triple-throw crank shaft, and each valve is worked by a rocking lever on the trunnion. Fig. 731 is a section through one valve box. Valve A is reciprocated by the trunnion lever, while valve B, used for reversing purposes, may for the present be considered fixed. C is the pressure supply, D the exhaust pipe, and E F the connection to the cylinder. Taking present position of B, a right-hand movement of A admits pressure to E, and a leftward movement permits exhaust from E, through H and G, to D. Supposing, now, B's position be so changed that H is opposite D, and G opposite F; the conditions are reversed, and a leftward movement of A admits pressure to F, while a rightward movement exhausts through the valve to D. Thus B is a reversing valve, and is moved by the piston of an auxiliary cylinder.

The Relief Valve J is simply a small, spring-loaded safety valve, which permits an escape of water whenever the pressure exceeds the normal, by reason of water inertia. Such valves are placed wherever there is liability to shock.

The Brotherhood-Hastie hydraulic engine, Fig, 732, is a combination of the well-known Brotherhood engine, p. 632, with Hastie's automatic stroke adjustment. Pressure water entering at P, passes to the cylinder by pipe A, and the exhaust returns through the same pipe, but is diverted by valve D into the outlet E. If P and E are connected to a reversing valve, the pressure water may enter at E and leave at P, and the direction of engine rotation





Working Valve.

Fig. 730.

Multiplying Gear for Hydraulic Cranes.

Fig. 729.

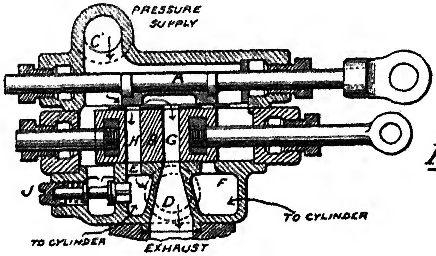
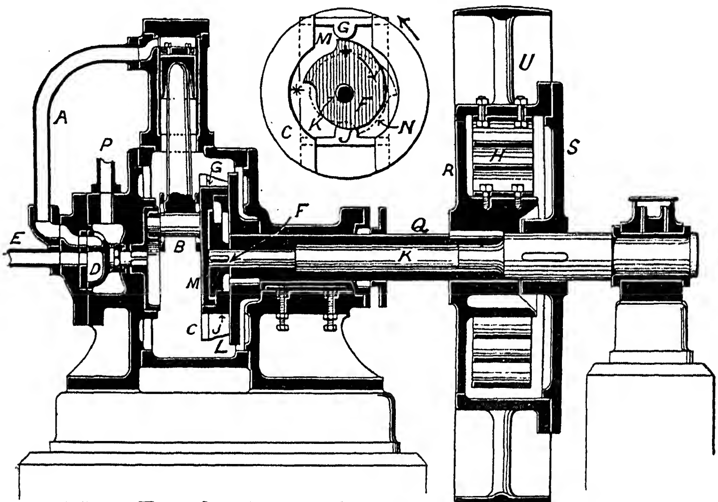


Fig. 731.

Armstrong's Reversing Valve.

is then reversed. The principal feature in this engine is the crank pin B, which is not fixed, but capable of sliding to a limited amount, within a diametral groove in the crank plate C, being for this purpose screwed into a shoe plate M. The power given to C is transmitted by a hollow shaft Q, through the strong volute spring H, to the driving pulley U. Now U is keyed to the inner shaft K, and when the load comes on the pulley there is a further coiling of spring H, which causes shaft K to turn relatively to G,



The Brotherhood-Hastie Hydraulic Engine *Fig. 732.*

through an angle depending on the turning moment. The result of K's turn is to rotate a cam F in such a way as to move the crank pin further from the shaft centre, and thus increase the throw; while on the other hand a decrease in load reverses the cam movement and enables the piston pressure to shorten the crank centres. Now there are two ways of accommodating fluid pressure to work required: alteration of stroke or of pressure. In the steam engine reduced work is met by reduced pressure; but water, being inexpansible, can only be adjusted in supply by a

corresponding adjustment of stroke. The result is roughly the same, for pressure × stroke = work done.

The cam F is peculiar in shape; it is shewn under full load, having turned through three-quarters of a revolution, in a right-hand direction. Its highest point is at +, and its lowest at *, and when the load is removed, the cam turns leftward until the projection at * stops itself against the projection G. Similarly the cam shewn dotted serves when the engine rotation is reversed, the projection J being then acted upon. Both G and J are one with the shoe-plate M, but lie in different planes, so that the two cams, also in one piece, may rotate without interference.

NOTES AND EXTENSIONS.

APPENDIX I.

(SECOND EDITION.)

CHAPTER I.

P. 37. **Brass and Gun-metal Founding.**—With these alloys a good height of pouring head is required, so as to cause pressure and prevent porosity in the casting. An ample number of risers should also be provided to permit escape of air or gas, and thus avoid honey-combing. No blackening is used, but the mould is faced with very fine sand.

P. 42. **Steel Casting.**—It having been noted that small castings were more porous than large ones, the conclusion was arrived at that it was of the utmost importance to keep the metal at a very great heat until poured. The present method of preventing blow-holes is to add silico-spiegel (a combination of silicon and pure cast iron) to the ladle while pouring, the better plan being to apply it in the molten condition. If this substance be not thoroughly mixed we get 'hard' and 'soft' spots in the casting, the former being due to accumulations of the silicon. In order to keep the casting uniformly hot until the whole mould be filled, the pouring gate should be chosen at the heaviest portion of the casting; and small castings are preferably poured from a small converter to themselves, instead of from the refuse of a large open-hearth melting. By facing the mould, where required, with ferro-manganese or ferro-chromium, very hard surfaces are obtained in those places, the latter substance giving the hardest result.

Chapter III.

P. 85. **Phosphor Bronze.**—The proportions by weight are as follows:—

For ordinary uses	Copper ...	85
	Tin ...	15
	Phosphorus	0·5
		100·5
Ditto...	Copper ...	90
	Tin ...	9
	Phosphorus	·75
		99·75
Tough metal for piston rings and eccentric linings	Copper ...	93
	Phosphor-tin	7
		100
For bearings (heavy machinery)	Copper ...	85
	Phosphor-tin	15
		100

Chapter IV.

P. 102. **Welding.** — It has been shewn by Sir Thomas Wrightson that the phenomenon of welding is akin to that of *regelation*, or the sticking together of ice under pressure. To prove this an experiment was made which shewed a distinct decrease in temperature (amounting to 106° F at 2550° F.) during welding, a similar result being known to take place during regelation. This abstraction of heat is caused by the melting of the iron or ice in either case, and the consequent need of latent heat for the liquefaction.

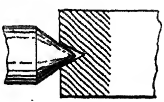
P. 124. **Case Hardening.**—If two pieces of iron, forming pin and socket respectively, are to be case-hardened, and a good *working fit* be finally required, it is important that the pin be made a pretty *tight fit* in the socket before hardening. After the

hardening process, both pieces will have swelled in volume, and it will be found that the pin will fit the socket more tightly than before. The final fit may be obtained by lapping the pin with emery powder in the lathe.

P. 128. **Hardening Steel.**—The difficulty experienced in hardening milling cutters without cracking is found to be largely due to unequal heating as well as unequal cooling. To avoid the former a method of heating in a bath of molten lead kept at a high temperature is found to be very successful. Regular cooling is very difficult to obtain in the case of thin articles, such as circular saws; but by placing a sheet of brown paper upon the surface of oil and allowing the article, placed upon the paper, to gradually sink into the liquid, warping may be largely prevented, though nothing softer than the equivalent of a brown colour can be thus obtained. If a saw is to be tempered to blue, the usual course of water tempering must be followed, dipping as smartly as possible, and the blade be straightened afterwards. Hardening in water and tempering subsequently in oil will produce a softer result than if water be used throughout.

Chapter V.

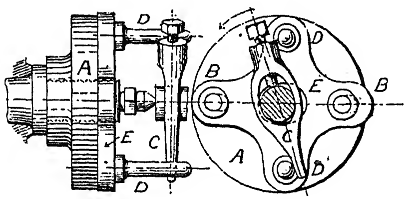
P. 152. **Lathe Centreing.**—By adopting a very slightly more acute apex angle for the centreing drill than for the lathe centre, the necessity for drilling the small hole, as mentioned at top of p. 152, is avoided. See Fig. 733.



Lathe Centre. Fig. 733.

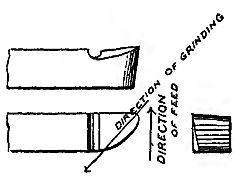
P. 153. **Double Driving.**—By allowing both ends of the carrier to be driven from the catch plate, stress is taken off the lathe centre, and more steady tooling is produced. Clements' driver, Fig. 734, is designed to effect this purpose. The carrier c

is driven by both the pins D D on opposite sides of it, and these pins are not fastened directly to the catch plate A, but to a separate cross-shaped plate E. The pins B B, holding plate E to the plate A, pass through slots in E, so that the latter is permitted to adjust itself to inequalities in the contour of the carrier C.



Clement's Double Driver Fig. 734.

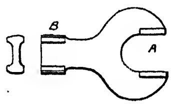
P. 157. **Lathe Tools.**—A simple and excellent roughing tool for a lathe is shewn in Fig. 735. A groove being fullered by the smith at a short distance from the tool point, the upper surface is ground to suit the tool angle in the direction shewn by the oblique arrow, while the relief angle is obtained by grinding both at front and side as shewn in the end view. (*See p.* 976.)



Lathe Roughing-Tool. Fig. 735.

Chapter VI.

P. 214. **Gauges.**—A very handy form of gauge is shewn in Fig. 736. It combines in one tool the equivalent of *plug gauge* B and *ring gauge* A. It is not quite so perfect in its application as the cylindrical gauges, but will serve most ordinary purposes.



Horseshoe Gauge Fig. 736.

P. 230. **Turning Eccentric Sheave.**—Where much of this work is to be done, and where the eccentricity of the sheaves to be turned varies much, the mandrel shewn at Fig. 737 is an ingenious and useful appliance. B is an expanding mandrel, having a cone A and four inclined keys D D that can be advanced outward or inward by the nut G, so as to grip the sheave firmly. The eccentricity may be adjusted by applying the handle H to the screw at B, and by unloosing and refastening the nut F. The whole is placed in the lathe by bolting the frame C to the face plate J, then advancing the poppet head to the centre F.

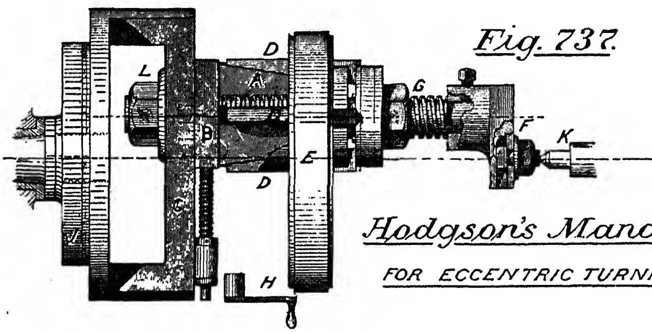


Fig. 737.

Hodgson's Mandrel

FOR ECCENTRIC TURNING

P. 232. **Boring Eccentric Straps.**—Instead of turning the straps in the lathe a heavy drilling machine may be conveniently adapted for boring purposes, as in Fig. 738. A large cutter C is placed horizontally through a slot in the boring bar F (or drilling machine spindle), and the radial feed may be obtained by giving a slight turn to the pinion D, which fits into teeth upon the cutter. B shews the eccentric straps, which are firmly bolted to the table A.

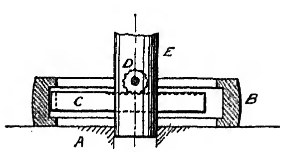


Fig. 738.

Boring Eccentric Straps.

P. 232. **Planing Eccentric Straps.**—The method shewn at E_1, Fig. 245, may be varied by using a shaping machine and

[illegible]ing the work to the side of the table, as at E, Fig. 260.

P. 235 *and* 243. **Machining Brasses.**—There are three methods of machining brasses for bearings and rods: (1) where the pair of brasses are cast in one, then bored, turned, and shaped, and finally cut through the middle with a parting tool; the space left by the tool is to be filled with a liner of the proper width. (2) The half brasses being cast separately are united with soft solder, and after tooling are separated by heating: here the mated surfaces must be first shaped. (3) Bolting together as described at p. 243.

P. 247. **Boring Crosshead.**—In case there should not be a bell chuck large enough for the purpose shewn in Fig. 254, the crosshead may be bolted to an angle plate placed on the face plate; but the setting is not quite so satisfactory.

P. 249. **Milling a Radius Link.**—The appliance shewn in Fig. 739 is for the purpose of guiding the link under the milling tool in such a manner as to cause a slot to be milled having the correct curvature. A and B are two slides hinged at C, and laid as shewn (plan view) upon the milling machine table. If these are now bolted down so as to enclose a very wide angle ACB, a large curve is the result, and if the angle be decreased, a smaller curve is traced on the link. EE are two dies having pins which carry the upper plate D, and on the latter the radius link is bolted in the position shewn. The right-hand die can be advanced by the screw F, this advance being the feed, and a very little thought will shew that the milling tool will cut out a curved slot whose radius will depend upon the angle ACB. (*See p.* 819.)

Leeds' Link-milling Apparatus.

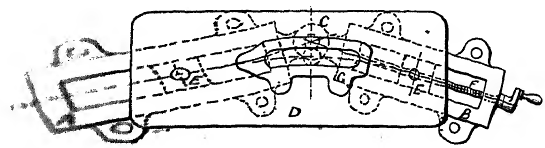
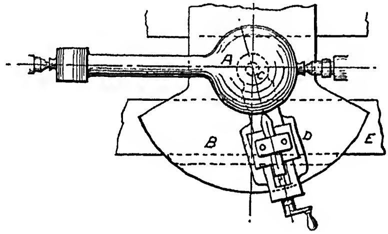


Fig. 739.

P. 256. **Turning Governor Balls.**—The method already described is adopted only where the special appliance shewn in Fig. 740 is not readily obtainable. The slide D, with its tool holder, is known as a ball-rest. It is pivoted at C, directly under the centre of the ball to be turned, and is supported upon the saddle B. The tool being set to turn the ball of correct diameter, the necessary radial feed is given by hand. The feed is more regular if spur teeth be cast upon the curve E, into which a pinion on the rest engages. A horizontal hand wheel fits then on the pinion spindle.



Ball Turning

Fig. 740.

P. 256. **Cutting Bevel-wheel Teeth.**—If not too small these wheels may be correctly cut by the machine in Fig. 741. The bars D D, carrying sliding tool boxes C C, are centred at A on a universal joint, and as the tools reciprocate a slow conical feed is obtained by guiding the bars round the plate on 'form' E, which is cut to twice the tooth scale and set at twice the distance from A. (*See pp.* 821 *and* 986.)

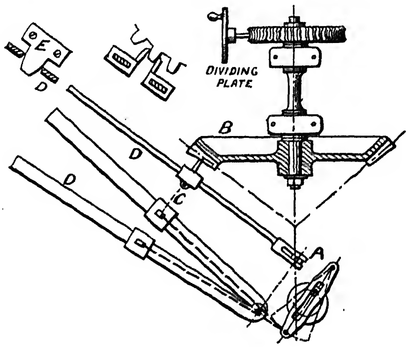
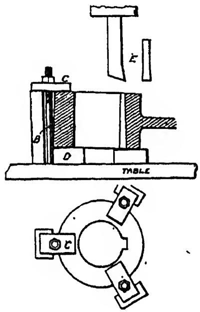


Fig. 741

Bevel-tooth cutting.

BY THE AID OF A FORM OR TEMPLATE

Cutting Key-ways in Wheel Bosses.—Usually the wheel is for this purpose laid horizontally on the table of a slotting machine, as in Fig. 742; and if taper be required, the boss is set at an inclination of 1 in 64 by packing at D. The clamps C and bolts D then hold the work securely. The cutting tool is sketched at F. It is now the custom in some shops to make the key-ways and keys perfectly parallel, but a very good fit with each other: a bursting pressure is thus avoided.



Cutting a Key-way, IN THE SLOTTING MACHINE

Fig. 742.

CHAPTER VII.

P. 289. **Hydraulic *v.* Electrical Transmission.**—In the discussion on the President's address before the N. E. Coast Institution of Engineers and Shipbuilders, October 16th, 1894, Mr. Tweddell says: 'Any remarks on the economy due to hydraulic transmission apply with equal force to electrical transmission. As the laws affecting both systems are almost identical, the question resolves itself into a matter of suitability for certain tasks, and as hydraulic pressure is suitable for intermittent work or for rectilinear motions, and not so suitable for rotary motions, it follows that a *combination* of the two systems . . . . is exactly what is wanted.'

P. 328. **Electric Welding.**—An interesting paper was read before the same Institution on February 12th, 1895, by Mr. Henry Foster, in which he described the methods adopted at the Newburn steel works. The 'Benardos' process was used, in principle

as shewn at p. 329, and the work to be welded consisted of general machinery repairs (especially for engine breakdowns) and boiler repairs. In addition the arc was used for boring holes in plates, or for otherwise melting portions of metal away. The process is of great advantage for breakdowns, as putting the machinery in working order in an extremely short space of time, and is also especially useful for patching purposes, thus saving many articles from the scrap heap. The average ratio of weld to solid was 85·5 per cent. for iron, and 80·8 for steel, as shewn by testing. The best hand welds were found to be much below this.

P. 693. **Fusible Plugs** made of gun-metal are screwed into the crown of boiler fire-boxes or furnaces. They are drilled through the centre and filled with a fusible metal whose composition depends on the temperature at which it is desired the steam shall blow out the fire when shortness of water occurs. The following table will shew the composition required:—

Melting Temperature in degrees F.	Composition of Metal in parts by Weight. Lead.	Tin.	Bismuth.
201	1	1	4
202	5	3	8
202	2	3	5
246	1	4	5
257	1	0	1
266	1	1	0
334	0	2	1
334	1	3	0
392	0	3	1

(*See also App. II., p.* 833.)

Chapter VIII.

P. 365–6. **Nature of Shear Stress.**

$$F_c = \sqrt{2}\,F_s \quad \text{and} \quad F_t = \sqrt{2}\,F_s$$

But F_c and F_t each act on areas of $\sqrt{2}$ while F_s acts on an area $= 1$ (*see Fig.* 324):

$$F_c = f_c\sqrt{2} \qquad F_t = f_t\sqrt{2} \qquad F_s = f_s$$

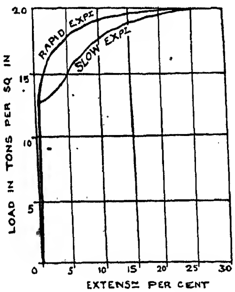
whence by substitution,

$$f_c\sqrt{2} = f_s\sqrt{2} \qquad f_t\sqrt{2} = f_s\sqrt{2}$$

$$\therefore f_c = f_t = f_s$$

or, a shear stress produces two normal stresses, tensile and compressive respectively, each of an equal intensity with its own, and having directions at 45° to the original stress.

P. 385. **Influence of Time on the Stress-strain Diagram.**—The diagram in Fig. 743, taken by Professor Ewing,



Influence of Time in Stress-strain Experiments.

Fig. 743.

shews very clearly the different plastic lines obtained, according to whether the test experiment be made very quickly or very slowly. It is clearly important that an average rate be maintained in applying the straining load.

P. 390. **Wöhler's Experiments.**—To give a further interest to Wöhler's important experiments, the three machines which he used are shewn in Figs. 744, 745, and 746. In the first an axle is loaded by a spring at a considerable distance 'over-neck,' and is then rotated several millions of times before breaking, the case being that of alternate stresses. The second machine, Fig. 745, represents the bending of a beam. The load is again caused by a spring in tension, and the varying stress is obtained by the rotation of the lever B, placed below. This is the case of a live load (removed and replaced). The third drawing,

Fig. 746, shews a tension experiment. In every case A is the specimen, B the rotating shaft, and C the load spring.

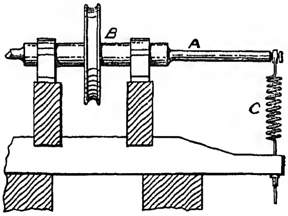


Fig. 744.

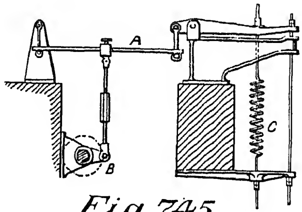


Fig. 745.

Wöhler's Experiments on Stress Repetition.

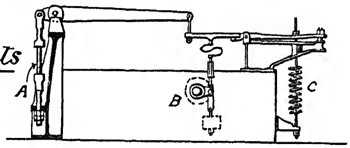


Fig. 746.

P. 399. **Thick Cylinders.** — The stresses within the material of a thick hollow cylinder are really somewhat complicated. They always consist of (1) a radial pressure, greatest at the inner circumference and decreasing to nothing at the outer circumference; and (2) a hoop tension, which is greatest at the inner ring and least at the outer ring, in the manner shewn in Fig. 352. This diagram shews what the hoop stress would be in an initially unstrained cylinder.

Now, in order to shew how changes in mechanical construction can decrease the hoop stress, and therefore *decrease the thickness of cylinder necessary*, we will put out of the question the radial compressive stress and the possible longitudinal tension, and consider only the *hoop stress* as tending to break the cylinder. Just as A B, Fig. 747, is the diagram of tensile hoop stress, C D is that for the fluid pressure, or *load*, and as A B = C D for conditions of strength,

$$pr = ft$$

as with thin cylinders, only that f is the *average* hoop stress.

Now, in this diagram it is evident that only the inner rings are of much value in resisting the load, and the outside rings do not

do anything like their share of the work, so to speak. We pass, therefore, to the practical principle of

Initial Stressing (during manufacture), by which the outer rings may be made of greater assistance. To do this, they are to be stressed initially in *tension*, and the inner rings in *compression*, and it will be shewn that when the fluid pressure is admitted, the internal hoop compression will be more than relieved, while the external hoop tension will be but slightly increased.

The first and most important method of arriving at the above result is to build the cylinder in separate concentric tubes, to be shrunk, the outer over the inner ones.

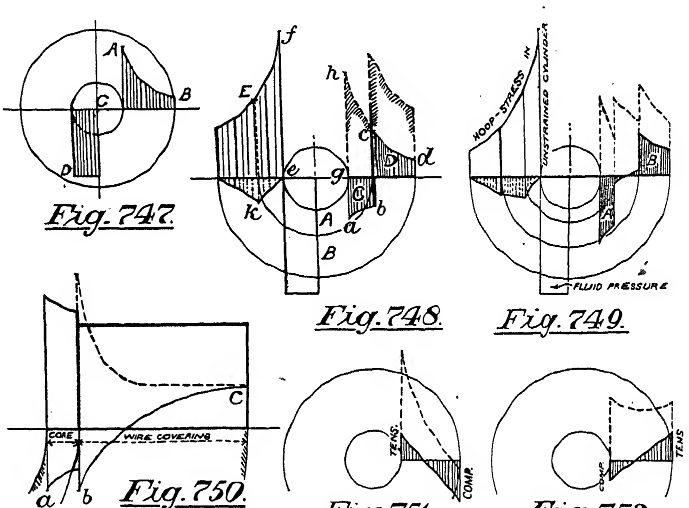


Fig. 747. *Fig. 748.* *Fig. 749.* *Fig. 750.* *Fig. 751.* *Fig. 752.*

Stresses in Thick Cylinders.

The tube B, Fig. 748, being shrunk over A, exerts a pressure between their common surfaces, as shewn on the left at *k*, and this pressure decreases to nothing at the inside and outside of the cylinder. But such pressure may be likened to a fluid pressure on the inside of B and the outside of A, and it will be easily

seen that hoop-stress diagrams like C and D will be formed, which can be calculated if we know the pressure between the surfaces. It must be noted also that C is compressive and D is tensile hoop stress. When, however, the fluid pressure is admitted to the cylinder, it will tend to equalise the stresses, for, supposing E to be the diagram obtained with a cylinder not initially strained, the diagram E must, in the actual case supposed, be superposed on C and D, having regard to sign, or be set up on the base line *a b c d*. The dotted areas will be the final result, and the real point for us to notice is that, instead of stressing our cylinder to *e f*, it is only stressed to *g h*, the outer rings taking their share, and thus the *thickness of the cylinder may be much less* than if it had not been initially strained. The areas C and D will be exactly equal, because there is equilibrium at first.

A still more equable stress may be obtained (and consequently less thickness required) by adopting a greater number of rings.

Fig. 749 is a diagram of the stresses when three rings are used, and will be easily understood from the last diagram. As before, A = B in area.

In applying these principles to the manufacture of large guns, both the initial and maximum firing tensions are kept within 18 tons per square inch. By using the modulus of elasticity it will be quite easy to find the hoop stress produced at the ring between the tubes, and conversely the radial pressure there, caused by extension due to shrinkage.

When guns are constructed by winding wire very tightly round a thin core, a perfectly equable stress may be obtained, and consequently these guns are lightest of all. In Fig. 750, dotted lines shew tension put in the wire as it is being wound on. The curved line *a b c* shews resulting stresses in the wire after the gun is finished, and the thick line the hoop stresses when the gun is fired.

Cast Iron Cylinders, if cast without any precaution, will be in a state of compression on the *outside*, after cooling, and of *tension* on the inside. Building then the hoop-stress diagram, Fig. 751, upon the incline, we get a very much worse result than before; for the initial stresses, caused by the inside cooling last (*see p.* 69), only assist the destruction of the cylinder when the

fluid pressure is admitted. We therefore require an *inordinate thickness.*

But if we cause the core, during casting, to lose its heat more rapidly than does the exterior, by circulating cold water through a central pipe, the inside will cool first, and a *tension* will be caused on the outside while a *compressive* hoop stress is felt on the inner rings.

Building now the 'fluid' hoop stress on the new inclined line, we obtain the dotted curve shewn in Fig. 752 The hoop stress is generally more equalised, and the internal hoop stress (or maximum) is decreased by the value of the initial compression. Again, therefore, we have been enabled to decrease the stress, and consequently the *necessary thickness*, by the artifice of initial stressing, caused by casting with a cold-water core.

Building-up from separâte tubes, or by tube and wire, must, however, be looked on as the very best methods, where, in fact, an almost certain desired result may be obtained. (*See App. II., p.* 841.)

P. 407. **Pitch of Riveting.**—It should be distinctly understood that the formula, pitch $= 1{\cdot}09 + d_1$ is only true if we use steel both for rivets and plates, and that we do not adopt plates above an inch in thickness. All other cases must be referred to the general formula on p. 407. The diameter of the rivet for a particular plate has, up to the present time, been fixed by practical considerations of punching. It is open to question, now that boilers are drilled, whether some alteration may not with advantage be made, tending to increase the diameter of the rivet for thin plates. (*See Appendix III., p.* 921.)

Pp. 442-6. **Position of Supports for Least Bending Moment.**—Let the beam A B, Fig. 753, be loaded uniformly, and let it be required to find the position of the two supports when the least bending moment shall act upon the beam. This will occur when the maximum stress caused by the load between C and D is equal to that caused by the loads at A C and D B. Thus, if we considered only the loads at A C and D B, a uniform bending moment is caused by them between C and D, and this moment has to be completely balanced by that at the

centre of the beam, caused by the load C D. In short, as $B_m \propto f \propto \delta$, M_1 must equal M_2, in order to cause the least stress in the beam. Assuming the lettered dimensions to represent

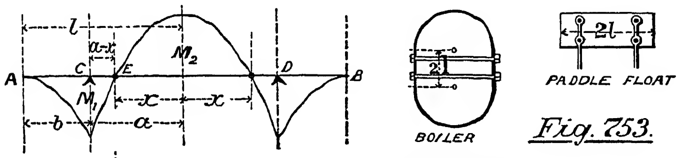


Fig. 753.

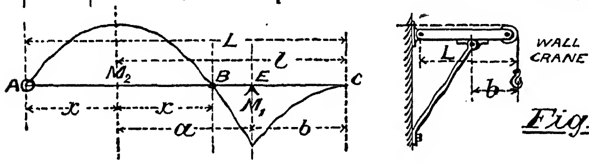


Fig. 754.

feet, and the load to be 1 lb. per foot run, then the bending moments are as follows:—

(1) Due to distributed load at $b = \dfrac{b^2}{2}$

(2) Due to distributed load at $a - x = \dfrac{(a-x)^2}{2}$

(3) Due to concentrated load at E $= x(a-x)$
(See also Fig. 400.)

(4) Due to distributed load at $2x = \dfrac{(2x)^2}{8} = \dfrac{x^2}{2}$

But $(2+3) = M_1$ and $(1) = M_1$
also $(4) = M_2$

$$\therefore\quad (4) = (1),\qquad \frac{x^2}{2} = \frac{b^2}{2},\qquad \text{and } x = b$$

Again $(4) = (2+3)$

$$\therefore\quad \frac{x^2}{2} = \frac{(a-x)^2}{2} + x(a-x)$$

$$2x^2 = a^2 \quad\text{or}\quad 2b^2 = a^2$$

$$\therefore\quad b = \frac{a}{\sqrt{2}} = \frac{l-b}{\sqrt{2}} \qquad \text{and } b = \frac{l}{1+\sqrt{2}} = \cdot 414\, l$$

There are two practical applications of this problem to which reference may be made. The first is where a locomotive boiler-barrel is sometimes flattened to save width, and the flat portion becomes such a beam as we have discussed, the supports being then represented by stay bolts. The other example is that of a paddle-wheel float, when supported by the float arms at two places only.

An extension of the problem is shewn in Fig. 754. There are two supports, A and E, but the former is a hinge. Using similar letters,

$$x = b, \qquad b = \cdot 414 l, \qquad l = L - x = L - b$$

$$\therefore \; b = \cdot 414 (L - b), \qquad \text{and} \quad \underline{b = \cdot 292\, L.}$$

A similar case, but with concentrated load at C, is that of the wall crane, Fig. 754.

P. 445. **Continuous Beams.**—The following is a simple method of finding the reactions on the supports in the case of a beam over two spans.

Suppose the mid-support be removed, a deflection will be caused by the *uniform* load, $2\,W_1$.

$$\Delta_1 = \frac{(2\,W_1)\,l^3}{48\,EI} \cdot \frac{5}{8} \qquad \text{(See pp. 451 and 849.)}$$

Now the upward pressure that would neutralise this deflection would have the same value as a *concentrated* load capable of causing it. Let this load = R: then the deflection caused by R,

$$\Delta_2 = \frac{R\,l^3}{48\,EI} \quad \text{and equating the two values,}$$

$$\frac{R\,l^3}{48\,EI} = \frac{(2\,W_1)\,l^3}{48\,EI} \cdot \frac{5}{8} \qquad \text{whence } \underline{R = \frac{10}{8}\,W_1}$$

CHAPTER IX.

P. 481. **The Lever-loaded Safety Valve.** The effect of the weights of lever and valve may be taken into account as a downward force at the valve centre, which has to be overcome by the steam pressure. In addition to the lengths shewn in Fig. 438 (measured in inches), and the weight W (in lbs.), Let

w = weight of lever, in lbs.

v = weight of valve and valve centre, in lbs.

l = distance from fulcrum to centre of gravity of lever, in inches.

p = pressure per square inch of the steam.

d = diameter of valve, in inches.

Then, Upward moments = downward moments

$$\left(\frac{p\,\pi\,d^2}{4} - v\right)a = \mathrm{W\,A} + w\,l$$

$$\therefore \mathrm{W} = \frac{(\cdot 785\,p d^2 - v)\,a - w\,l}{\mathrm{A}}$$

P. 487. **The Quadric Chain.** Referring to Fig. 449, the examples at 3 and 4 are spoken of as *parallel-crank* chains; while that at 1 is called a *lever-crank* chain, from the fact that the beam rocks like a simple lever, and the crank completely rotates.

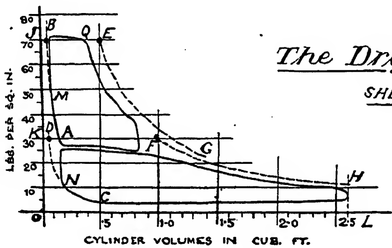
P. 503. **Oldham's Coupling,** Fig. 474, is derived from the elliptic trammels, Fig. 448, where the trammel bar is fixed and the cross revolves. The relative motion being unchanged, a fixed pencil held against C's face will produce an elliptical arc. Also C's centre will describe a circle passing through centres A and B (*Euclid*, III. 31), for the angle between grooves is always a right angle. Hence (*Euclid*, I. 32) the angular movements of A and B are identical.

P. 512. **Approach and Recess.** During approach the flank of the driving wheel is acting on the point of the driven wheel (*see* Fig. 501), while during recess the point of the driver acts upon the flank of the driven wheel. On the supposition that the *pushing* friction during approach is more prejudicial than the

trailing friction during recess, some clock makers cut away the flanks of the driving wheel and the points of the driven wheel. A much larger number of teeth are then, however, necessary to secure smooth action.

CHAPTER X.

Pp. 594 and 620. **The Dryness of Steam.**—The proportionate dryness of steam at all points of the stroke can be very conveniently represented upon the indicator diagram by means of a simple construction. Imagine the indicator cards in Fig. 755 to have been obtained in the usual manner, and then to have been plotted by the method shewn on p. 622, the clearance volumes being of course known in terms of the cylinder volumes. Let it also be supposed that the weight of steam passing through the cylinders at every stroke has been found experimentally by measuring the feed water entering the boiler, the latter being of course only occupied in supplying steam to the engine. Referring



The Dryness of Steam
SHEWN GRAPHICALLY.

Fig. 755.

next to the diagram on p. 598 or to the table of saturated steam volumes in this appendix, the volume of 1 lb. weight at 70 lbs. absolute pressure is found to be 6·1 cub. ft., from which the volume of steam due to the feed water weight can be at once deduced. Similarly the same quantity of steam, when expanded to 30 lbs. absolute pressure, will have a volume that may be ascertained from the tables, or can be found by constructing the saturation curve.

Referring again to the diagram, the compression curves are to be continued, as at A B and C D. Then at B E set off the steam

volume at 70 lbs. pressure, and at D F that due to 30 lbs. pressure. Through E and F respectively draw the saturation curves E G and F H, which for rough purposes may be hyperbolas with origin O; but, for greater accuracy, may be drawn with a new origin for each curve, according to the manner described on p. 624, and shewn in Fig. 624. Thus J E being divided into 6·1 parts, and K F into 13·5 parts to find the scale of specific volume, the origin for each curve must be moved ·41 ft. to the right and ·35 lb. below the old origin, and the curves are then treated as hyperbolas from the new origins. The method of constructing an hyperbola is shewn at p. 615.

It will be seen that there is a separate saturation curve for each cylinder of the compound engine, but this is only due to the fact that the clearance volume in the L.P. cylinder does not bear the same relation to the steam volume as did that in the H.P. cylinder. If it be desired to shew both cards under one saturation curve, it is only necessary to transfer the lower card to tracing paper and move it to the right until the curves E G and F H coincide. Or another method is to re-set out the cards by placing the curved lines C N D and A M B upon the vertical line O K J, and so move the cards to the left, distorting them somewhat. One saturation curve may then be drawn through E, B E being the same as before, for the clearance steam is now eliminated. Of course, instead of completing the compression curves by the dotted lines, the curves could have been drawn for any pressures between A M in the H.P. cylinder, and between C N in the L.P. cylinder.

Assuming now that the cushion steam is dry—an assumption involving but very slight error—the dryness of the steam may be found for any point in the stroke of either piston, by comparing the total volume of steam in the cylinder with that shewn by the saturation curve at that point, the difference of these volumes shewing the amount of steam condensed by the cold cylinder walls. Thus, at commencement of the H.P. stroke:—

$$\text{Dryness fraction} = \frac{\text{J Q}}{\text{J E}}$$

$$\text{Wetness fraction} = \frac{\text{Q E}}{\text{J E}}$$

And these quantities may be ascertained for other points of the stroke by measuring similar intercepts along horizontal lines.

It will be noticed that the steam always tends to become drier toward the end of the stroke (due to re-evaporation), though never becoming entirely so; and the value of compounding in drying the steam is also apparent. (*See Appendix II., p.* 878.)

P. 608. **Adiabatic of Saturated Steam.**—Zeuner gives the following empirical formula to find the value of the exponent n in the equation to the adiabatic for saturated steam :—

$$n = 1{\cdot}035 + (\cdot 1 \times \text{dryness fraction}).$$

from which it would appear that Rankine's curve is suitable for steam having 25 per cent. of suspended moisture, while Zeuner's curve is for steam initially dry.

P. 609. **Expansion of Wet Saturated Steam.**—It has been already pointed out that the saturation curve is a curve of lowering temperature. Also that the compression of dry saturated steam at constant temperature causes it to become wet by partial condensation, because steam at a given temperature can only exist at a certain pressure. Conversely, the expansion of steam at constant temperature will tend to dry it; but if there be sufficient water present, the pressure will not fall during the expansion, and therefore the curve on the diagram would be a horizontal straight line.

Calculation of Specific Volume of Dry Saturated Steam, when the temperature, pressure, and latent heat are known.

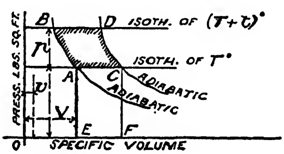


Fig. 756.

Dealing with 1 lb. weight of steam, whose volume and pressure are shewn at A, Fig. 756:

Let L = latent heat at temp. T in foot-pounds;

T = temp. of steam along ac, absolute;

t = small difference of temperature between steam at ac and that at mn;

s = volume of 1 lb. of water near boiling point in cub. ft.;

V = volume of 1 lb. of steam at temp. T in cub. ft.;

P = absolute pressure at ac [illegible] per sq. ft.;

p = small difference of pressure between ac and mn [illegible] per sq. ft.;

H = heat supplied while expanding along ac, causing work area [illegible] (foot-pounds);

h = small heat supplied due to temp. increase t, causing small work area [illegible] (foot-pounds).

[illegible] represents increase of volume as steam, V − s the increase of volume from water to steam. But these increments are [illegible] caused by the heats H and L. Therefore,

$$H = L \quad [illegible] \quad \text{or } H = \frac{L \times AC}{V - s} \qquad (1)$$

Again, it will be seen on reference to a temperature entropy diagram (see *Appendix II*, p. [illegible]) that

$$H = [illegible] \text{ and } h = \frac{Ht}{T} = \frac{L \times AC}{V - s} \times \frac{t}{T} \qquad (2)$$

But amnc = ac × p, and the work area thus formed is equal to the heat supplied for the purpose, or

$$h = AC \times p \qquad (3)$$

Equating (2) and (3), $AC \times p = \frac{L \times AC}{V - s} \times \frac{t}{T}$

Specific vol. $V = \frac{Lt}{Tp} + s$

In applying this formula, take t = 1°, and from the table find the corresponding increase of pressure p in lbs. per sq. ft. Take the volume s of 1 lb. of water = [illegible] cub. ft. Inserting also the absolute temperature T and latent heat in foot-pounds, the specific volume is found. (See *Appendix III*, p. 933.)

P. [illegible] Engine and Boiler Efficiencies.—There are very many ways of stating the economy of a steam engine, which,

if not made quite clear, are calculated to confuse the student. We will, therefore, explain each method as fully as need be.

1. *Duty.*—This is the oldest method. The boiler and engine are considered as one machine, and the number of *foot-pounds of work* done are stated, as obtained from 1 *cwt. of coal*, the best Welsh coal being generally used. The value of this quantity having been found from time to time for the best engines (about 60 to 100 million foot-pounds), constituted a sort of standard for the performance of others.

2. *Coal burnt per I.H.P. per hour.*—This, the later standard, also including both boiler and engine, exists more or less at the present time. Its value was, at one period, as much as 4, but even with single engines it was soon lowered to 3, in two-stage compounds to $2\frac{1}{2}$ or 2, and in triple-stage compounds to 1·5 or even 1·3. The connection between standards 1 and 2 is shewn as follows :—

$$\text{Duty} = \frac{112 \times 33{,}000 \times 60}{\text{lbs. of coal per I. H. P. hr.}}$$

a relation easily proved by simple proportion.

3. *Steam used per I. H. P. per hour.* — This is a standard of comparatively recent introduction, and is adopted where the engine's performance is to be gauged apart from that of the boiler. The feed water weight is measured, and priming eliminated as much as possible. The value may vary from $12\frac{1}{2}$ lbs. in extremely good cases to 30 lbs. with ordinary working; 24 being good for non-condensing engines.

4. *Efficiency of Boiler.*—Just as the weight of steam used is often taken to represent the efficiency of the engine, so the weight of water evaporated is often called the efficiency of the boiler. The evaporation per lb. of coal may be taken at 10 lbs. in good cases. A better statement of boiler efficiency may be made by separately calculating the heat units given to the water and the heat units obtained by the combustion of 1 lb. of coal. Then,

$$\text{Boiler efficiency} = \frac{\text{Heat units given to water}}{\text{Heat units from coal}}$$

which may have a value of 70 per cent. in a good boiler. (*See p.* 1004.)

5. *Efficiency of a Perfect Engine*, that is, of an engine having a reversible cycle. This is the highest efficiency to be obtained by an engine working between given temperatures, and may therefore be termed the *ideal efficiency.* It has already been shewn to be

$$\frac{\tau_1 - \tau_2}{\tau_1}$$

where τ_1 is the temperature of the furnace and τ_2 that of the condenser. It includes, therefore, the efficiency of the boiler, but, dealing only with the diagram horse power, does not take account of mechanical imperfections. Supposing such efficiency once obtained with given temperatures, the engine might yet be improved by increasing the range of temperature, provided we do not exceed the bounds of reason. Very often this efficiency is taken for the engine only, with τ_1 as live steam temperature; but the steam being at a much lower temperature than the furnace, a much lower ideal efficiency is possible. A very good value by the first method is 77 per cent., and by the second, 32 per cent.

6. *Thermal Efficiency of a Real Engine.* — There are two methods of stating this, the relation of the work done to the heat expended, according to whether we include the boiler and engine, or take the engine alone. Thus we have :

$$\text{Thermal efficiency } (a) = \frac{33{,}000 \div 772}{\text{heat units in steam per I. H. P.}}$$

$$\text{Thermal efficiency } (b) = \frac{33{,}000 \div 772}{\text{heat units from coal per I. H. P.}}$$

Then a good value for *a* might be 14 to 20 per cent., while for *b* it might be 9 to 13 per cent. Of course, if the thermal efficiency is to be compared with the Carnot cycle, the latter must be measured by similar temperatures. (*See p.* 883.)

7. *Relative Efficiency.*—This, as its name implies, is a comparison of the thermal efficiency of an engine with that of a reversible cycle having the same range of temperature, and is the

fairest method of showing an engine's performance. It may be stated thus:—

$$\text{Relative efficiency} = \frac{\text{Thermal efficiency}}{\text{Efficiency of Carnot cycle}}$$

Of course, whatever is included under the thermal efficiency will be included in the Carnot efficiency as well. The value of the relative efficiency may vary from about [illegible] per cent. (*See Appendix II., p. 885.*)

8. *Mechanical efficiency of the engine* merely takes account of engine friction and compares the work on the indicator diagram with that given to the crank shaft. Then,

$$\text{Mechanical efficiency} = \frac{\text{Brake horse power}}{\text{Indicated horse power}}$$

And a value of 85 per cent. is considered good.

9. *Total efficiency of a real engine.* Lastly, the total percentage of work gained, at the shafting, by the burning of a given quantity of coal may be stated as

(I.) Boiler effy. × thermal effy. [illegible] × mechanical effy.

But, as we can never get more work out of the coal than is shown by the ideal efficiency, the fairest statement is as follows:

(II.) Boiler effy. × relative effy. [illegible] × mechanical effy.,

from which we may at once find whether our engine and boiler are wasteful. (*See also Appendix II., p. 885, and Appendix IV., p. 966.*)

(*P. 612.*) **Reversible Cycles.**—Any engine that transforms all the heat, represented by drop of temperature, into work, is a reversible engine, and the arguments on p. 611 fully apply. The thermal efficiency of such an engine would be the same as Carnot's (p. 611).

Now there are two methods of arriving at a reversible cycle. The first method is shown by the Carnot cycle, where heat is only taken in at the highest temperature, and rejected at the lowest temperature. Were intermediate temperatures adopted for some of the heat, such heat would have less "availability," and as proved that no heat is supplied or removed between t and t_0 the curves

of expansion and compression between these temperatures are strict adiabatics.

The second and only* other method of obtaining reversibility is to use a *regenerator*, as first practically attempted by Stirling, and afterwards by Ericsson in their respective air engines. Diagram A, Fig. 757, shews Stirling's cycle. The gas having a pressure

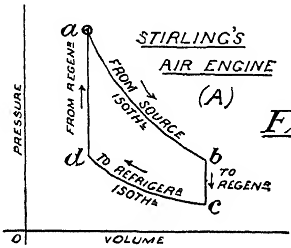
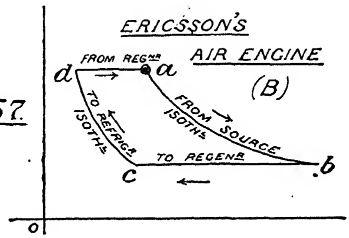


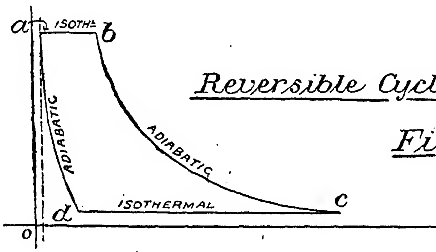
Fig. 757.



and volume corresponding to *a*, took in heat along the isothermal *a b*; rejected a portion during *b c* to the regenerator, at constant volume; was compressed isothermally from *c* to *d*, during which time heat was rejected to a refrigerator; and between *d* and *a* again received the heat which was rejected from *b* to *c*. Although *b c* and *d a* are substituted for adiabatics, the giving and receiving of heat is strictly within the engine itself, the heat rejected at *b c* being fully returned at *d a*, so the reversibility is unimpaired. Diagram B illustrates the cycle of Ericsson, which is only different from that of Stirling in that the regenerator gives or abstracts heat at constant pressure. As before, *a* may be considered the starting-point. * (*Excepting as in Appendix II.*, *p.* 883.)

P. 613. **Reversibility in the Steam Engine.** — It appears, therefore, that supply and rejection must be along isothermals, and expansion and compression along adiabatics (unless a regenerator be used). Remembering that isothermals for saturated steam are horizontal straight lines, the reversible cycle in Fig. 758 for the steam engine is easily understood. Thus *a b* is the isothermal of reception, *b c* the adiabatic of expansion, *c d* the isothermal of rejection, and *d a* the adiabatic

of compression. Such an engine uses the same substance over and over again, and the cycle is strictly reversible, for we have complete expansion to *c*, and complete compression to *a*. Its



Reversible Cycle for Steam.

Fig. 758.

efficiency therefore is measured like the cycle of Carnot. The substance is water at *a*, and steam and water at other times.

P. 623. **Ratio of Expansion in a Single Cylinder.**—As a result of many recent practical experiments, it is found that a ratio of expansion of between 6 and 9 is the highest limit which can be adopted with economy, though a ratio of 4 will yield practically as good results.

P. 624. **Theoretical Diagram.**—In order to make the calculations from the preliminary diagram agree with practical diagrams, Prof. Unwin introduces a fractional coefficient in the formulæ, which he calls the *diagram factor*, and which is deduced from experimental results of engines similar to that under consideration. The value of the factor varies from ·85 to ·95 in good engines.

P. 637. **Exhaust Lap.**—In the case of a D valve, an early cut-off to steam will cause an early cut-off to exhaust (*see* Zeuner's diagram, p. 660). In such a case, if a later compression point be advisable, it may be necessary not only to eliminate exhaust lap entirely, but actually to give a small opening to exhaust when the valve is at mid stroke. Such opening is then termed *negative lap*, and would be shewn on the steam circle in Fig. 653.

P. 650. **An Isochronous Governor** is a governor having but one speed consistent with stability. A parabolic governor is isochronous excepting for the influence of friction, its stable speed occurring when the balls are in their lowest position. If this speed be increased the equilibrium is neutral, because the height of the cone does not change with the rise of the balls. All over-sensitive governors *hunt* more or less, that is, they are apt to rise too high and fall too low, even with the small changes of load during a revolution, and the result is a condition of oscillation which prevents a settled position corresponding to engine speed. This extreme sensitiveness can be prevented by a spring (Fig. 645), but in a much better way by a dashpot (Fig. 261), for in the latter case the air causes a constant though small resistance.

P. 700. **Ignition Tubes.**—Porcelain tubes have been used for some time (1895), their life being about twelve months. Iridio-platinum is also now adopted (1898) with great success.

Description of an Engine for Refrigerating Air.—When work is done on a gas, without subtraction of heat, the whole work is expended in raising the temperature of the gas; and when a gas does work without addition of heat the whole work is obtained by the abstraction of heat from the gas, causing a decrease of temperature. In practice these results are only partially effected, but the general changes obtained may be understood by reference to Fig. 562, and P. 547, where (theoretically) air at 60° F. is cooled to −201° F.

The work derived from the expansion of steam in the engine cylinder is employed to drive the piston of an air-compressing cylinder, which draws air on the inner and compresses it on the outer stroke, up to 90 or 100 lbs by gauge. The temperature rising to 200° or 300°, the cylinder is jacketed with cold circulating water, to prevent damage to lubricants, and thus the air is *partially* cooled. Leaving this cylinder, it passes through the pipes of a surface condenser, being there cooled to 60° or 70° by cold water circulation round the tubes; and from the condenser it enters an expansion cylinder covered with a non-conductor. Here the air does work, helping the engine, and, cooling to

-10° or -30°, is exhausted to the storage chamber, from which, after doing duty, it is redrawn by the engine to supply the compression cylinder. By this arrangement there is less heat to abstract than if air at 60° were used, and an important economy results. The compression and expansion cylinders must lie apart from the steam cylinder and from each other, and all the pistons are connected to one crank shaft, which has, usually, a heavy fly wheel. The parts required are therefore, in order:—Steam cylinder, compression cylinder with water jacket, condenser, expansion cylinder (non-conducting), and storage chamber (non-conducting).

At first sight it would appear that heat abstraction was entirely due to the use of condenser and water jacket, but this is only a part of the truth. Heat given during adiabatic compression

$$= \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} = \frac{c}{\gamma - 1} \times (\text{diff. of temp.})$$

$$\left.\begin{array}{r} \text{But } \gamma = \dfrac{K_p}{K_v} \\ \text{and } K_p - K_v = c \end{array}\right\} \therefore \frac{c}{\gamma - 1} = (K_v \times \text{diff. of temp.})$$

Again, condenser and jacket remove heat at constant pressure, and the amount

$$= K_p \times (\text{diff. of temp.})$$

Also, heat removed during adiabatic expansion

$$= K_v \times (\text{diff. of temp.})$$

Assuming, as in the theoretical example, that rise of temp. during compression = fall of temp. during expansion, the various heats may be represented as follows.—

$$N = \text{normal heat (atmospheric temperature)}$$

1. After compression, heat in gas

$$= N + K_v \quad \text{(latter given by engine)}$$

2. After condensation, heat in gas

$$= N + K_v - K_p \text{ (heat now in condenser} = K_p)$$

3. After expansion, heat in gas

$$= N + K_v - K_p - K_v$$
$$= N - K_p \quad \text{(heat taken by engine} = K_v)$$

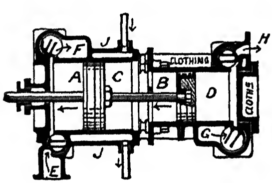
Let condenser heat go to boiler, and thence to engine; let all expansion heat be abstracted by engine; and let the total, together with remaining heat in storage chamber, be used to compress the gas; then

$$\begin{array}{lrl} \text{From storage chamber...... Heat} & = & N - K_p \\ \text{,, condenser and jacket ,,} & = & K_p \\ \text{,, expansion cylinder... ,,} & = & K_v \\ \hline \text{Total} & = & N + K_v \end{array}$$

which is the same as (1), or the cycle is complete, and the action (if without loss) would go on for ever.

Taking values for K_p and K_v and N as normal heat; the deficiency in N after condensation is represented by $K_p - K_v = 53{\cdot}2$, and a further deficiency, shewn by $K_v = 130$, occurs during expansion, the heat abstractions being therefore 29 per cent. and 71 per cent. of the total abstractions, respectively. If the drop be less than the rise of temperature, the condenser will abstract a larger proportion, and if the condenser could take its heat at constant volume, the expansion would do the whole work of heat abstraction, for then the condenser abstraction would equal the compression supply. The condenser water goes partly to feed the boiler, and partly to heat work-rooms, &c.

Fig. 759 is a section through the air cylinders, whose pistons are connected to the engine crank-shaft. A, the compression



Refrigerating Engine.

Fig. 759.

cylinder, receives air from the storage chamber, through E, which after compression escapes by F to the condenser. Thence it re-appears, cooled to 60° or 70°, and entering the expansion cylinder D by passage G, the piston moves upward. The air volume having shrunk by cooling, cylinder D is much smaller

than [illegible] exhausted, [illegible] the storage chamber [illegible] chambers [illegible] of air [illegible] separate, and the pistons are packed with [illegible] hard on their outer circumference [illegible] with rubber rings [illegible] secure air tightness.

The arrangement of Fig. [illegible] being duplicated constitutes a double-acting [illegible], or double-acting cylinders may be placed side by side [illegible]

Table of the Properties of Saturated Steam

Absolute Pressure, *p*	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
0 85	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
5	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1 5	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
2 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
2 5	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
3 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
3 5	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
4 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
4 5	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
5 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
6 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
7 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
8 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
9 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
10 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
11 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
12 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
13 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
14 0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
14 7	212	[illegible]	[illegible]	[illegible]	[illegible]
15 0	213	[illegible]	[illegible]	[illegible]	[illegible]
16 0	216	[illegible]	[illegible]	[illegible]	[illegible]

Table of the Properties of Saturated Steam (continued)

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
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APPENDIX II.

(THIRD EDITION.)

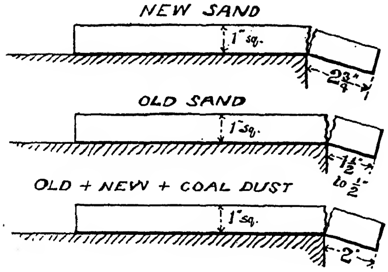
CHAPTER I.

Pp. 3 and 42. **Remelting of Cast Iron.**—Experiments made by Fairbairn in 1853, by melting Eglinton hot-blast pig up to eighteen times, shewed that while at first there seemed to be some improvement, there was a deterioration in the later meltings, the iron becoming white and hard. Chemical examinations then made by Snelus, and lately repeated by Mr. T. Turner on the original pieces (lent by Prof. Unwin) indicate that the action is one of oxidation, resembling that of the puddling furnace or Bessemer converter.

MR. THOS. TURNER'S ANALYSIS.

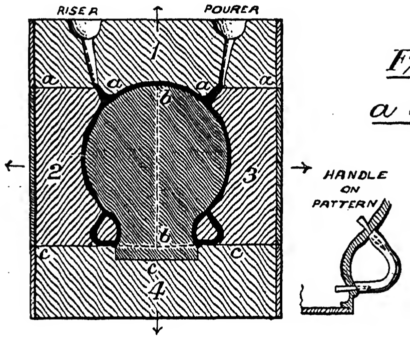
Number of melting.	Total Carbon.	Combined Carbon.	Silicon.	Sulphur.	Mangan.	Phosp.
1	2·67	·25	4·22	·03	1·75	·47
8	2·97	·08	3·21	·05	·58	·53
12	2·94	·85	2·52	·11	·33	·55
14	2·98	1·31	2·18	·13	·23	·56
15	2·87	1·75	1·95	·16	·17	·58
16	2·88	2·00	1·88	·20	·12	·61

P. 5. **Moulding Sand.**—*Floor sand* may have 6 parts by weight of old sand to 2 of new sand and half a part of coal dust; *facing sand*, 6 of old sand to 4 of new sand and one of coal dust. Too much burnt sand, even if ground up, cakes when re-used, and causes the metal to boil. Too much wetting is as bad as too little, but the requisite consistency may be roughly tested by grasping a handful of the sand, which should just retain its shape when the hand is again opened. A more scientific method is used by Mr. Bagshaw, who prepares bars of sand in moulds, by light pressure, 12 inches long and 1 inch square, which he slowly pro-



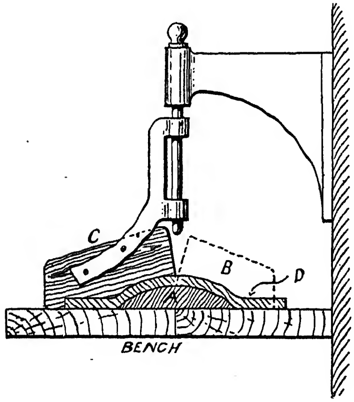
Sand Testing.

Fig. 760.



Four-part Box.
a three-legged pot.

Fig. 761.



Making a
Plaster Pattern

Fig. 762.

jects over a table, as in Fig. 760, till they break by the weight of the over-hanging portions, the results being seen in the diagram. Unless the sand be properly prepared by riddling, treading, and wetting, no amount of venting can make a good casting.

P. 12. **Four-part Box.**—However intricate in form a casting may have to be, the difficulties of moulding may always be met by the introduction of sufficient boxes or of loose pieces. Fig. 761 shews a method of moulding a three-legged pot or 'skillet,' practised at the Carron ironworks for an almost unknown period. Four boxes are used, numbered 1, 2, 3, and 4, which give also the order of removal, there being partings at *a a*, *b b*, and *c c*. The handles are pegged loosely to the pattern from the inside, and are afterwards removed in a downward direction, the core being struck on a rough iron body.

As evidence of what may be done when given enough moulding-boxes and loose pieces, there was exhibited at Paris in 1889, by the Société Cockerill, a single casting of 10 tons weight, representing three marine cylinders, with standards, bed-plate, feed- and air-pumps.

P. 14. **Plaster Patterns.**—The use of these is not difficult to explain, their introduction here being due to their great similarity to loam patterns. Fig. 762 will indicate the process, it being desired to make a plaster pattern for a dome cover. Firstly, a supporting mound A is made by the rotation of board B over moist plaster-of-Paris, the vertical spindle hanging from a wall bracket. When dry, the mound is painted with shellac varnish, and the thickness piece described by the board C. The pattern D, thus formed, is afterwards removed, varnished, and used exactly as a wooden pattern would be; and with care may serve for a large number of impressions.

A further use of plaster occurs in moulding thin flat objects. Imagine a flat cover, Fig. 762*a*, say for a sand-box. Firstly, a wooden pattern, being impressed in sand as at A, and the parting made, the upper box is filled with plaster. When the plaster sets, the sand is removed, the plaster surface varnished; and the bottom box similarly treated, as at B. Secondly, the boxes are separated and the pattern removed. Thirdly, a new pair of boxes is provided of the same dimensions, and, taking each box sepa-

rately, a plaster *cast* is taken from each of the plaster *blocks* just made. Now these *casts*, after drying, could not be fitted together, for each is larger than its respective *block* by the thickness of the casting; but if each *cast* be separately impressed in sand, and the

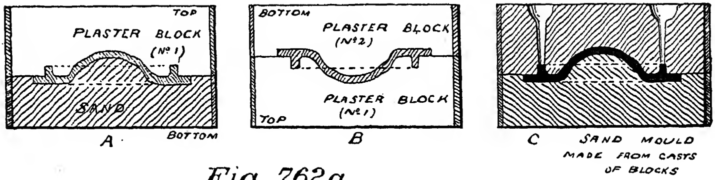


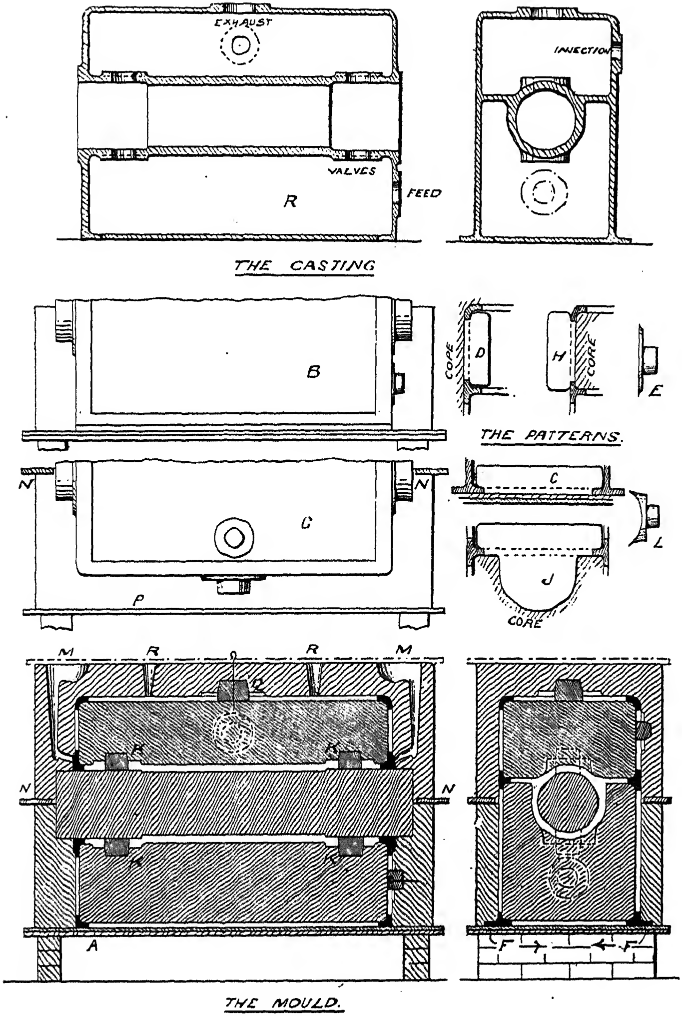
Fig. 762a.

Flat Moulding with Plaster Blocks.

two sand impressions fitted together, they will appear as at C, the thickness space being left, into which the metal is run.

The casts having been varnished, will materially increase the speed of moulding, for a pair of casts may be divided between two men, and the removal of the cast from the sand is more expeditious than a thin pattern. The blocks may be retained for future casts, but are of no use in moulding.

P. 27. **Cubical Moulds in Loam.**—Deeming it advisable to illustrate flat mouldings in loam by at least one example, the jet condenser, Fig. 763, has been chosen. The casting R consists of a rectangular box containing a pump barrel, and having suitable openings for exhaust, feed, and injection. It was formerly usual to make complete patterns for such objects, but skeletons are now largely adopted, the flat intermediate spaces being struck by loam boards. In our case the wooden skeleton is seen in position in the mould, being there shewn by the dense black portions, and in describing its use we shall begin with the cope mould. An iron plate A is placed on brick supports, and a coating of loam laid on, which is then smoothed over with a plain board. The pattern being set upon this bed, right side up, as at B, loam is filled in round the side; then, by using suitable striking-boards, as at D and C, the various flat surfaces are finished in facing loam, and a pattern E embedded for the feed print. The skeleton is now removed vertically, but the bottom strips F F, and the feed flange,



Moulding a Condenser in Loam.

Fig. 763.

being loose, are afterwards removed horizontally. The top half of the pattern is similarly treated, by embedding in loam as at G, and the plates N and P are bolted together to facilitate lifting and turning over. Here there are two loose flanges as at E, and the runner and riser patterns, all of which have to be inserted.

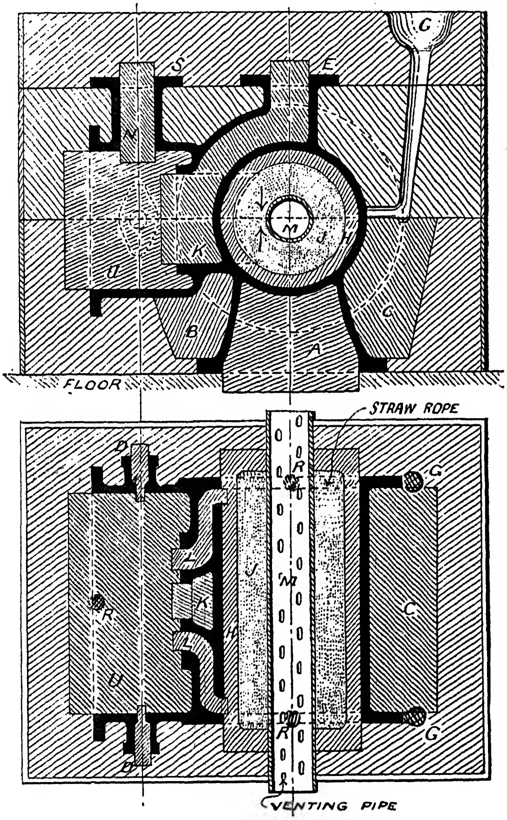
The two halves of the main core are next made, the skeleton being again used; but this time the loam boards are swept outside it as at H, while the pump barrel is struck by the board J for the lower half, and by a similar, but shallower, board for the upper half. The flanges are supplied by pattern L. The two main cores are, of course, made separately, and in order to remove them the guiding strips for board J must be only temporarily fixed, and the skeleton itself must also be split at the same place. The cylinder core is struck on an iron barrel (see p. 14), and cores are provided for the holes K K, Q, &c.

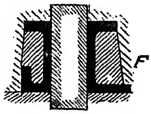
Lastly, the various parts of the mould are put together: first the lower cope, then the lower core, the cylinder core, upper core, and the upper cope, inserting the before-mentioned short cores, and thus making ready for casting.

P. 31. **Steam Cylinder in Greensand.**—Cylinders of the smaller size cannot be moulded in loam like that on p. 21, so a short account is here given of a greensand mould, aided by Fig. 764. The pattern is split horizontally through the centres of steam chest and barrel, and is supplied with prints B and C to secure withdrawal of the lower half, the sand pockets A, B, and C, being afterwards filled by cores made in suitable boxes. A core U is also moulded for the steam chest, and the barrel core is struck on a pipe M, as already described at p. 14; J showing the straw rope, and H the loam covering. Examining the direction of withdrawal of the upper half of the pattern, shewn by the arrow. the steam and exhaust-pipe flanges are seen to be troublesome parts. This is a case where some care in design would obviate much after expense, for the flanges could easily be made to draw if the pipes were equal to them in diameter; as shewn however, they are loose on the pattern, and a third box is provided for their subsequent withdrawal. One other method, at F, requires ring cores between flange and cylinder body. The port cores L L, pipe cores N and K, and stuffing-box cores D D,

complete the mould, which, finally supplied with gates and runners G G and risers R R, is ready for casting.

P. 34. **Moulding Machines.**—Besides the machine on p. 32, which has only one use, that of moulding wheel teeth, there

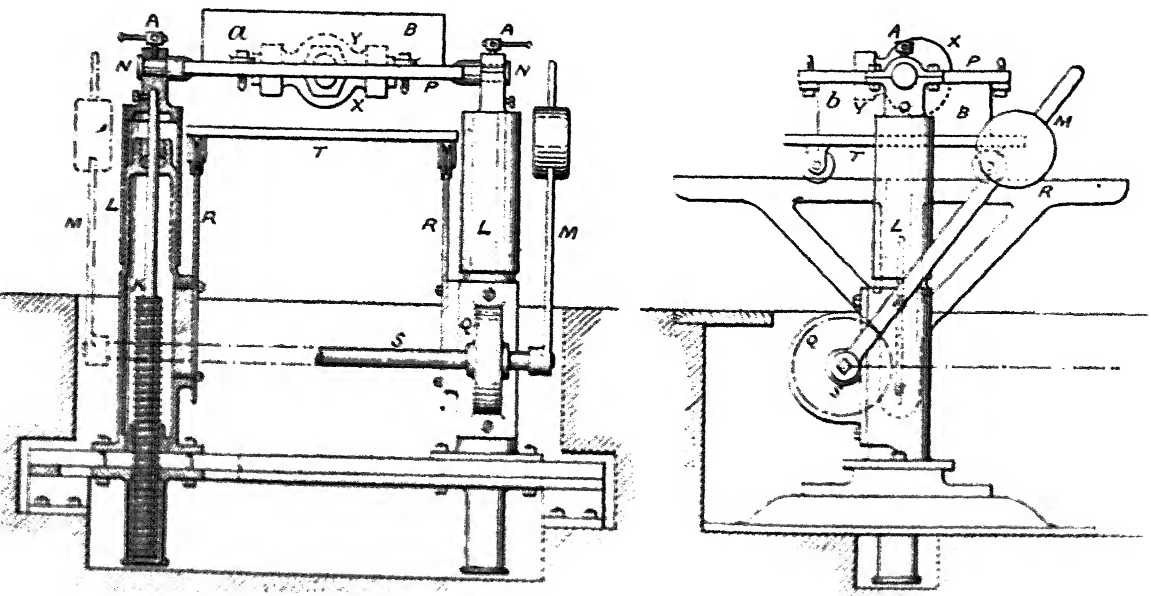




Moulding a Steam Cylinder in Greensand

Fig. 764.

are many machines suited to general repetition work, which is thus done both more quickly and more accurately. The operations of hand moulding are more or less simulated, that is, the pattern is first secured in position and fastened to the box, the sand is next rammed, usually with the box upside down, and



Woolnough & Dehne's Moulding Machine Fig 765

lastly the pattern is withdrawn *upward* or the box *downward*. Woolnough and Dehne's machine, Fig. 765, provides for a raising of the pattern, a turning over, and removal of box on a short tramway. R R are the rails, clamped at a convenient height, T a table on wheels, P a pattern plate, turning over when required, and at other times clamped horizontally by screws A A. The pattern halves X and Y are first screwed to this plate in mutual correspondence, and the raising and lowering is performed by levers M M, which act on pinions Q Q through shaft S, thereby moving racks K K. The upper ends of the racks support sleeves L L, which carry trunnions N N, thus lifted or depressed as required.

The operations can now be understood. Assuming the box on the top of the plate as at *a*, it is filled with sand and rammed, the screws A A and the cotter bolts preventing rotation and lifting respectively. These screws are next released, the plate raised, and the box turned through 180° into position *b*, an intermediate raising being necessary. Lastly, the cotters are withdrawn, the plate raised, and the box removed by the tramway. This leaves the X half of pattern uppermost, and the previous operations being repeated for it also, the boxes are bolted together for casting.

To avoid the lost time due to raising and turning over, Mr. J. Maclellan has devised a machine where P is rigid, and the box, being always right side up, is filled with sand and lifted till it meets the pattern. The ramming is then performed hydraulically, by the raising of a second box of sand, which is pressed against the first one, thus squeezing some of its contents through the ribs and producing the necessary consistency. (*See p.* 969.)

P. 42. **Whitworth Compressed Steel.**—See p. 790.

CHAPTER II.

P. 44. **Woods.**—Passing inward through a tree section, one meets in order the bark, sapwood, heartwood, and pith. The heartwood is best, and the sapwood should be avoided if possible.

The circular marks are called *annular rings*, while the radial ones are termed *medullary rays*, and the process of drying tends to split the wood along the latter, as already shewn. To minimise this fault the tree should be cut into balks before drying, and the last should preferably be done naturally and gradually, over some two or three years, during which time it is protected from rain, but allowed free air-current. Artificial drying or desiccation produces more splits, or 'shakes' as they are called, and of the latter, 'cup' shakes follow the rings, and 'star' shakes the rays, but the worst shakes are those that twist as they travel along the log. Speaking generally, timber comes under one or other of two great divisions—the pine wood, and the non-resinous or leaf wood. Under the former we have all the softer woods, such as pines, firs, and spruces; while the latter includes the hard woods, such as oak, beech, elm, sycamore, ash, mahogany, &c.

CHAPTER III.

P. 74. **The Blast Furnace.**—Before iron ore is smelted it is often *calcined* or heated alone, either in open heaps or in kilns, one heap of 2000 tons being kept hot for about three weeks. Some ores, such as the Scotch, require no fuel for this purpose, they themselves containing free carbon. An interesting method is adopted in Styria, where the ore travels slowly down an inclined kiln, the fire heat passing in the reverse direction. With the use of the hot blast, calcining is not so necessary.

The design of the blast furnace depends largely on the district, that on p. 73 being from Cleveland, where poor ores and coke fuel are the rule, and the dimensions large. In Scotland, where rich ores and coal fuel are used, even a 60-feet furnace is almost too high, causing much trouble in keeping up the fuel at the boshes. A large furnace may have an output of as much as 500 tons per week. The blast air is heated to about 900° Fahr. by the waste gases from the top of the furnace, and the nature of the charge will depend very materially on the ore, that for Dowlais

clay ironstone being (according to Bloxam) for every ton of iron smelted—

Grey Cast Iron	Calcined Ore	48 cwt
	Coal	50
	Limestone	17
White Cast Iron	Calcined Ore	28 ,,
	Hematite...	10 ,,
	Forge Slag	10 ,,
	Coal	42 ,,
	Limestone	14 ,,

and one-third more air is supplied in the second case. Hematite is a very pure red ore containing some 70% of iron, while the above-mentioned ore will never have more than 50%.

P. 74. **Cast Iron (Effect of Elements).**

Carbon exercises the greatest change in cast iron, and its effects have been well explained.

Silicon, after carbon, is the most useful element. If present up to 3½% it produces soft, strong, grey iron; and if added to a hard, whitish, and cheap iron, it will make it strong and grey. It is now much used to improve poor irons, but should only be present in small quantities in iron that is to be chilled.

Sulphur is prejudicial, causing blowholes: it should not exceed ·15%.

Phosphorus is also harmful, for though giving fluidity, it produces brittleness if in excess of 1%.

Manganese tends to dissolve the graphite and promote combined carbon, and confers the property of chilling. If more than 1% it causes large crystals as in Spiegeleisen. Rapid solidification also favours combined carbon (see Chilling, p. 34).

The following foundry irons are the mean of many good specimens:

WOOLWICH, 1858.

	Graphite.	Si.	P.	S.	Mn.
Percentages	2·59	1·42	·39	·06	·58

ROSEBANK, SCOTLAND.

(Strong foundry iron.)

	Comb. C.	Si.	P.	S.	Mn.
Percentages	·46	1·31	·53	·06	·99

P. 80. **Spiegeleisen** owes its value mostly to the presence of manganese, which varies from $3\frac{1}{2}$ to $11\frac{1}{2}\%$ at different times, but less than 6% renders the material useless for steel making. There is also about 5% of combined carbon, and the fracture shews large crystals. Probably, too, the silicon present is of use, for both manganese and silicon generally improve the quality of steel.

P. 82. **Whitworth Compressed Steel.**—Apparently the existence of blowholes in steel ingots is due to the fact that low-carbon steel, when molten, is capable of occluding or holding in solution certain gases. When solidification sets in, these gases are liberated from the fluid only to be immediately imprisoned by the solidifying steel, while slower cooling only increases sponginess, for then more gases are given off. To avoid the loss occasioned by the cutting away of the ingot head, and to improve the rest of the metal, three principal methods are in vogue:

(1) Chemical treatment {addition of silicon or manganese}.
(2) Forging or Cogging {under steam hammer or hydraulic press}.
(3) Compression {when fluid}.

Silicon and manganese diminish the gas released, and collect such bubbles as are already formed, but they reduce ductility.

In the Whitworth process, a powerful hydraulic press is applied to the molten steel when in the ingot mould, producing a thoroughly sound material by the elimination of all bubbles. The press is shewn in Fig. 766. The fixed head H rests on four screwed columns DD, which are again supported on the base-casting A. The pressure water is admitted under ram B, which

rises against the trolley C supporting the ingot mould R; and the load is resisted by the head F, being transmitted thereto by the fixed plunger E. The head F is supported by rods PP attached to the lifting plungers NN, and its upward movement is prevented by the nuts GG, while through it there pass two rapid-pitched screws MM, upon each of which is a wheel L gearing with the nuts GG; and lastly there is a small hydraulic cylinder J, whose piston moves a rack in gear with the wheel K, which again forms a nut on the screw M. Supposing it be required to raise F to admit the mould, the piston J is moved so as to turn K, and with it M, thus releasing nuts GG, and moving them upward to a very small amount. Then piston J is locked in the new position. Next, the rams NN are raised, lifting F and also the screws MM through wheel K, which is now a fixed nut. MM thus revolving, the nuts GG are moved upward at the same rate as F. When the proper height has been reached, the plunger J is moved back to its original position, bringing nuts GG on to their seats to receive the upward thrust.

The ingot mould consists of iron rings SS, in two concentric sets, within which are placed blocks of firebrick TT, and a lining of ganister. At each end, ring plates UU are fixed, to hold the bricks in place, and the open mould is covered with loose pistons QQ, again protected by fireclay slabs. When compression occurs, the plunger enters the mould by the rising of the latter, and the gases that escape through the bricks pass upward or downward, finally leaving by the holes in plates UU. A very high intensity of pressure is absolutely necessary, less than 15 or 20 tons per square inch being very doubtful policy. The press shewn can exert a total pressure of 10,000 tons. (*See third preface.*)

P. 82. **The Basic Steel process,** known also as the 'Bessemer-Basic,' was introduced by Thomas and Gilchrist in 1886 for producing steel from phosphoric pig, which had previously proved useless for steel making. Its success is due to a magnesia lining to the converter, obtained by crushing dolomite or magnesium limestone that has been previously dried, mixing it with tar, ramming it as a lining, and heating to 'coke' the tar. When this 'basic' lining has been heated, 14 to 20 % of the charge weight is thrown in as burnt lime, after which the pig is

added, and the 'blow' proceeds till the iron is free of carbon; the 'after-blow' then takes place till the phosphorus is eliminated. The converter is now slightly tipped, some ferro-manganese

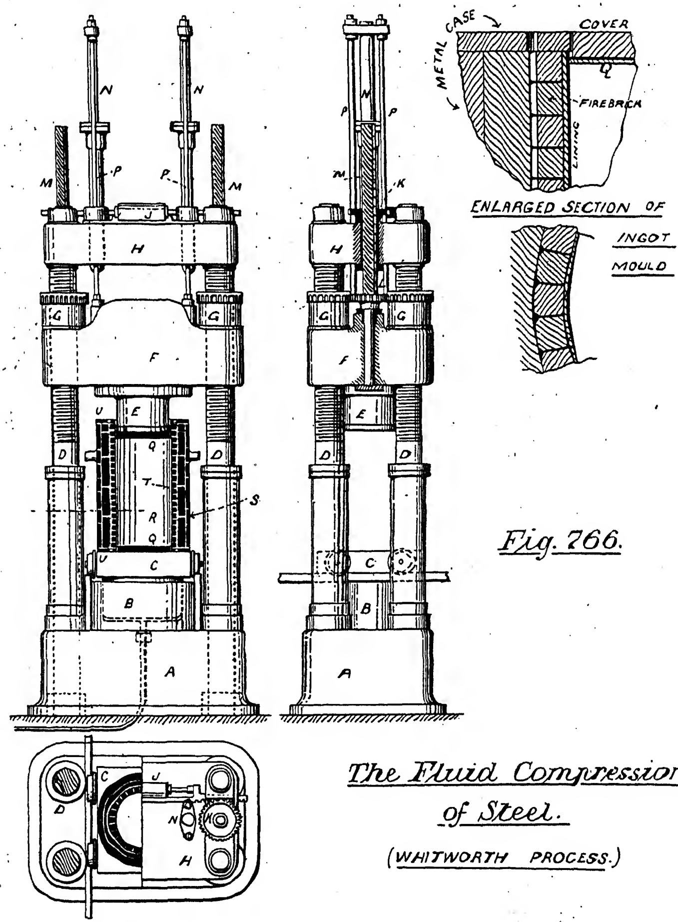


Fig. 766.

The Fluid Compression of Steel.

(WHITWORTH PROCESS.)

added, the slag removed, and the metal poured into ingots. The percentage of silicon and sulphur should be very low, and there should be from 1 to 2 % of manganese. If this be followed, the

resulting product, basic steel, is the nearest possible thing to metallic iron. Finally, in the

Acid-Bessemer Process	Silicon supplies heat for conversion	Phosphorus must be low
Basic-Bessemer Process	Phosphorus supplies heat	Silicon must be low
Open-Hearth Process.	Heat supplied artificially.	Suitable for intermediate pigs, with, say, ·2 to ·5% Phosphorus.

and the Basic process is chiefly of value on the Continent, where phosphoric ores abound.

P. 83. **Temper of Steel,** or the proportion of carbon to suit it to a particular purpose. The following table is the result of actual analyses of Siemens' steel, extending over some three or four years, and may be looked on as reliably representing present-day practice. It shows the gradual tendency to decrease carbon percentage, except in regard to steel used for cutting tools :

TEMPER OF STEEL.

		Percentage of Carbon.
Castings	Locomotive wheel centres, locomotive firebox girders, marine stern frames, &c.	·3 to ·4
Forgings	Crank pins, crank shafts, connecting rods	·25 to ·3
	Piston rods to stand wear	·35
Chains	To weld well	·15 to ·18
Springs—laminated		·4 to ·6
Boiler plates—ordinary		·17 to ·2
Boiler plates—for welding		·15 to ·17
Tool steel		1·7

The percentage in Bessemer steel is somewhat lower, if the same strength is to be retained. Wrought iron, also by analysis, contains from ·25 % of carbon down to mere traces.

P. 84. **Electro-deposited Copper.**—Ever since the discovery of electro-metallurgy it has been known that purer copper could thus be obtained, but the metal proved insufficiently dense, and very weak. These difficulties have been overcome by

Mr. Elmore's process, which is principally used for the making of pipes. An iron mandrel is placed in insulated bearings in a solution of copper sulphate, and a number of unrefined copper bars placed round it some distance off. The mandrel is connected to the negative pole, and the bars to the positive pole of a dynamo, and the copper is thus decomposed and deposited on the mandrel at a rate of about ·2 inch in 170 hours. At the same time a piece of polished agate presses on the mandrel, and travels slowly from end to end backward and forward, so as to cover the whole surface as the mandrel slowly revolves; the copper is therefore being burnished as fast as it is deposited, and is thereby made very dense and strong. When the pipe is sufficiently thick, the mandrel is taken off and steamed, which allows the pipe to expand so as to be easily removed. It is stated that copper pipes thus made are 50% stronger than those that are either brazed or solid-drawn, and have a superior ductility; while further strength can be imparted by rolling.

Manganese Steel is obtained by adding ferro-manganese to iron or low-carbon steel. The first attempt, with $2\frac{1}{2}$% Mn, at Terre Noire about 1885, resulting in a brittle metal, the experiments were abandoned; but later, Mr. Hadfield (1887), by pushing the percentage higher, obtained complete success. The results at various degrees are very curious, and are probably explained by the presence of carbon, which is inevitable.

Percentage of Mn.	Cast Manganese Steel.
$1\frac{1}{4}$	: Produces no change if C be low.
$3\frac{1}{2}$ to 5	: Remarkably brittle cold, even with only ·5% C, but not so when hot.
$5\frac{1}{2}$ to $6\frac{1}{2}$	: About the same.
$6\frac{1}{2}$ to 7	: Strength and ductility increased; magnetic quality decreasing.
9 to 10	: Very ductile.
10	: If not very tough, can be improved by water quenching. Too hard for filing. Strength equals crucible steel.
12	: Entirely lacking in strength, no matter what the treatment.
13	: Practically non-magnetic.
14	: Maximum strength.

There is a rapid decrease in strength with any further increase in manganese. It should be noted that the carbon must be kept down to 1% in the 14% material, to do which the ferro-manganese should have about 82% of Mn. No doubt if the carbon could be sufficiently decreased, even 20 or 25% Mn could be added. The ingots are from 28 to 30 cwts., and the ferro-manganese is added in a molten state. Honeycombing is not bad, but the centre of the ingot 'pipes' considerably on account of the great contraction of this steel.

FORGED MANGANESE STEEL
(Water quenched).

Percentage of Mn.	
10	: Ductility equal to mild steel, strength much greater.
13	: Still higher strength and ductility.
14	: Limit of manufacture: beyond this toughness decreases.

Finally, manganese steel is strong, ductile, and hard; free from blowholes, and more fluid than cast steel, but pipes badly, and requires good feeding gates to mould. Its hardness prevents machining or fitting, and grinding only can be adopted. The steel is suited to dredger pins and other articles subject to great wear; and is also useful for wheel tyres in conjunction with chilled brake blocks, causing great grip. The following is an analysis:

Carbon.	Silicon.	Manganese.
·85%	·28%	14·1%

and the strength and ductility are next shewn:

Percentage Mn.		Breaking Stress sq. in.	
$2\frac{1}{2}$ to $7\frac{1}{2}$ ..	Cast	.. $3\frac{1}{4}$ tons	.. very brittle
$2\frac{1}{2}$ to $7\frac{1}{2}$ ..	Forged	.. 25 tons	.. 3% elongation
14 ..	Water toughened	.. 56 to 65 tons	.. 38 to 50% elongation

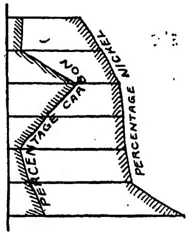
(as against Basic cast steel: 22 to 27 tons: 26 to 30% elongation).

In the testing machine the material is semi-plastic from the commencement, shewing permanent set with low loads.

Nickel Steel is a very valuable material, discovered about 1885, in the search for combined strength and ductility, which carbon alone is unable to give to iron. But when it is said that

1% of nickel increases the cost of the steel by one-third, the difficulties of introducing it commercially will be understood, so that though the percentage Ni might rise to 20 or 30 with advantage, the practical limit is 3 or $3\frac{1}{4}$, with which there is usually ·3 to ·4% carbon. The following list, from actual specimens, will shew that hardness is due to the presence of carbon:

Percentage Nickel.	Percentage Carbon.	Quality.
2·05	·22	soft
2·62	·19	soft
3·1	·96	very hard
3·2	·54	medium hard
3·25	·16	soft
3·4	·31	slightly hard
4·95	·51	medium hard



All these specimens welded well, and could be bent cold. As regards strength and ductility the following are from actual tests:

Use, &c.	Percentage Nickel.	Breaking stress tons sq. in.	Elastic limit tons sq. in.	Elongation per cent.
Boiler plates	3	35·7	22·3	25
Ship plates	3	37	26·7	20
No. 13 Ni Steel ...	2·05	37·8	—	31·5
Gun tube............	—	41·6	26	21·2
Gun jacket	—	44·6	26·8	20·4
No. 14 Ni Steel (for propeller shafts)	3·35	45	33·5	27·5
Gun hoops	—	48·7	30·4	20·5
Wire .... (easily drawn)	30	88·7	—	6·25

In other cases it has reached 40 tons breaking stress with 28 tons elastic limit, and an elongation equal to mild steel. Compare this with Forth Bridge steel at 30 tons breaking and 17 tons elastic, or the Eiffel Tower steel with 22 tons breaking and 16 tons elastic. In all cases the nickel steel has the same ductility as mild steel, but with 30% greater tenacity and 75% greater elastic

strength. It resists shock, is very uniform in structure, flanges well, and is less corrodible than mild steel. It is therefore suitable for shafts, propellers, ship plates, boiler plates, large guns, and lastly, when surface-hardened, for armour plates. Evidently the function of the nickel is to prevent the shortness caused by carbon, while permitting and even assisting the latter to exercise its strength-giving property; it is, however, unable to confer hardness without the assistance of the carbon, but increases that hardening capacity.

The Harvey Process is applied to armour plate to give it such extreme surface-hardness as will resist the attack of shell. The process is essentially one of part cementation, or the introduction of carbon to a given depth, and has been practised in England by laying the plate on a shallow fireclay box filled with charcoal, luting with fireclay, and keeping at 2400° Fahr. for several weeks. As the plates weighed 30 or 40 tons each, the risk of breaking the boxes was very great. This objection is removed at the Bethlehem Steel Works, U.S.A., where two plates are hardened simultaneously, face to face, but with 8 ins. of charcoal dust between. They are then placed on supports within a furnace, thickly luted with sand and fireclay, and gradually heated up to 1700° Fahr., remaining at that temperature for 8 or 10 days, after which they are taken out and laid on supports in an empty tank, so as to keep them apart, and allow water pipes to pass between and around them. From these pipes a spray of ice-cold water is directed on the plates for about an hour, and the final cooling is done in an oil tank. The oxide on the plate surface is afterwards removed by a pneumatic chipping-chisel (*p.* 949).

Armour plates are thus hardened to a depth of about $1\frac{3}{4}$ ins., and cannot be drilled or otherwise machined unless locally softened by an annealing process. For this purpose an electric current of large volume, from an alternating dynamo, is sent through the plate at the required place, heating it to 1000° Fahr.; and the temperature is then let down gradually to a dull red, which is tested by the burning of a pine stick in contact with the plate. The electrical principle involved is exactly the same as that in the Thomson process of welding (p. 329), the dynamo providing a current of 100 ampères at 300 volts, which is changed

to 10,000 ampères at the plate, passing there through copper terminals ½ in. square, kept cool by water circulation. The current is very gradually applied, and very slowly shut off, by means of a rheostatic switch.

The value of nickel steel for armour plates is shewn by the following figures, which prove a saving in weight of 43·8 % for equivalent resistance, over ordinary steel plates:—

Kind of Plate.	Relative penetration.	Relative resistance.
Soft Steel	2·20	·455
Compound Steel and Iron ...	1·75	·572
All Steel	1·64	·609
Nickel-Harvey	1·00	1·000

Chrome Steel is obtained by the addition of chromium to steel having about ·4% of carbon, and the resulting metal is not only extremely hard, but is 'self-hardening,' that is, it only needs to be cooled in a current of air after forging, to make it suitable for metal-cutting tools or armour-piercing shells. For the latter purpose the French Government use—

Carbon.	Chromium.	Nickel.
·4%	1%	2%

while other analyses give—

Carbon %	Chromium %	Manganese %	Silicon %	
·6	2·2	not known	not known	French.
·63	1·04	·05	·15	American.
·44	·92	·03	·12	American.

A 12 in. chrome-steel projectile has pierced a 16-in. armour plate, while a chilled chrome-steel armour plate has received a

17 in. projectile, fired with 8 cwt. of powder, the impression being but $1\frac{3}{4}$ in. deep. (*See p.* 976.)

Tungsten Steel (called also Mushet's Steel) is also a self-hardening steel, similarly suitable for projectiles and cutting tools, and having a composition by analysis of—

Carbon %	Tungsten %	Silicon %	Manganese %
1·36	2·58	·42	·25

(*See p.* 976.)

Sterro Metal is a brass to which have been added small proportions of iron and tin. Its percentage composition is—

Copper... ...	55 to 60	Iron	2 to 4
Zinc	34 to 44	Tin	1 to 2

It is both cheaper and stronger than gun-metal. Speaking generally, the presence of iron reduces the tenacity, but in the proportions shewn is of value. This metal has been used for hydraulic pumps.

Delta Metal, though its proportions are unpublished, seems to be simply Sterro metal in a forged or rolled condition. The mechanical treatment thus received considerably increases its strength, and it is much advocated for ship propellers. Though the presence of iron causes a slight rust, the loss after six months' immersion in an acid-impregnated water was but 1·2% as against 46% with wrought iron or steel.

Silicon Bronze is an alloy of copper and silicon, the latter acting as a flux or reducing agent, clarifying the copper and preventing oxide scale. The resulting product is very strong, as here shewn:

Percentage Copper.	Percentage Silicon.	Breaking stress, tons sq. in.	Elongation %
97	3	$24\frac{1}{2}$	55
95	5	$33\frac{1}{2}$	8

The second sample is deficient in toughness, and more than 5% silicon causes great brittleness. Adding ·1% of silicon to melted copper produces clean castings by removing oxide, and a little silicon to any brass or bronze is advisable. Silicon bronze

corrodes rather more than aluminium bronze, but is very close-grained, and therefore suitable for resisting fluid pressure.

Aluminium is procured from its ores by one or other of two processes. The older or Deville process has been much improved as the Castner process, and is thus practised at Oldbury, near Birmingham, being based on the displacement of aluminium from its ores by metallic sodium. Caustic soda and iron carbide are melted in furnaces at 1470° Fahr., and, thus being kept for some $1\frac{1}{2}$ hours, the *sodium* distils over into iron condensers, and is afterwards cast into blocks of 2 lbs. each. Some 20 furnaces are kept going at once, each producing 60 lbs. of sodium per day, with an expenditure of 360 lbs. caustic soda and 300 lbs. iron carbide.

Next, *alumina* is prepared, by mixing the ground mineral (bauxite) with soda ash, and heating it in a furnace till silicate and soda aluminates are formed, after washing which, first with water and then with hydrochloric acid, the hydrate of alumina remains. This is mixed with common salt and charcoal into a paste, and made into balls, which are thoroughly dried and heated in earthen cylinders. While in this condition perfectly dry chlorine gas is passed over them, and *aluminium bichloride* distils over.

Lastly, 80 lbs. of the bichloride, 25 lbs. of metallic sodium, and 30 lbs. of cryolite (another ore of aluminium) as a flux, are heated together to 1830° Fahr.; and metallic *aluminium* to the weight of 8 lbs. is thereby produced, impure only to the extent of 2 %.

In the Cowles process the ore is directly reduced in electric furnaces, or rectangular fireclay pits kept hot by the current from an enormous dynamo. Each pole within the furnace consists of a bundle of five 3-inch carbons, having metallic caps or heads—of iron if ferro-aluminium is required, and of copper if for aluminium bronze. The furnace lining is made of lime and charcoal powder, the latter for localising the heat and saving the furnace materials. The charge consists of ore, metal (copper or iron as desired), and charcoal; and the resulting alloys contain 15 to 17% of aluminium.

Aluminium has only a third the specific gravity of iron, and is practically untarnishable. The addition of $\frac{1}{2}$ to 1 % to cast iron

increases fluidity and makes casting possible with the whiter irons. (*See p.* 1014.)

Aluminium Bronze, as a substitute for gold, has been long known, but has only recently been advocated as an engineer's metal. The best proportions are 5 to 11 % of aluminium, the rest being copper; and the strengths are, with

5% Al.	24·5 tons sq. in., breaking.	40% elongation.
11% Al.	35·7 tons sq. in., breaking.	10% elongation.

If a small portion of silicon be added, the strength is increased but the ductility diminished. The 10 % alloy is much used for bearings, gear wheels, propellers, &c. Shrinkage when casting is very great, and good feeding gates are necessary.

Manganese Bronze has been mentioned at p. 85. The ferro-manganese usually added is objectionable as introducing iron, which decreases toughness and increases corrosion, so it is better to use an alloy of manganese and copper. One of the best and cheapest manganese bronzes has the following percentage proportions:

Copper	53	Manganese ...	3·75
Zinc...	42	Aluminium ...	1·25

In view of the competition between the bronzes for propeller construction, it may be noted that the relative cost for different metals is given by the subjoined figures:

Cast Iron ...	£24	Gun Metal ...	£130
Steel	£38	Manganese Bronze	£135
Delta Metal ...	£115	Aluminium Bronze	£145

Phosphor Bronze ... £170.

CHAPTER IV

P. 98. **Steam Hammer Blow.**—For the benefit of some readers an amplification is here given of the matter on p. 98:—

Velocity due to tup weight $= \sqrt{2gH}$ (H = height of drop).

$$\text{Acceleration due to steam} = \frac{\text{Total pressure}}{\text{mass}} = \frac{Pg}{w}.$$

$\therefore$ Velocity due to steam $= \sqrt{2fH} = \sqrt{2\frac{Pg}{w}H}$

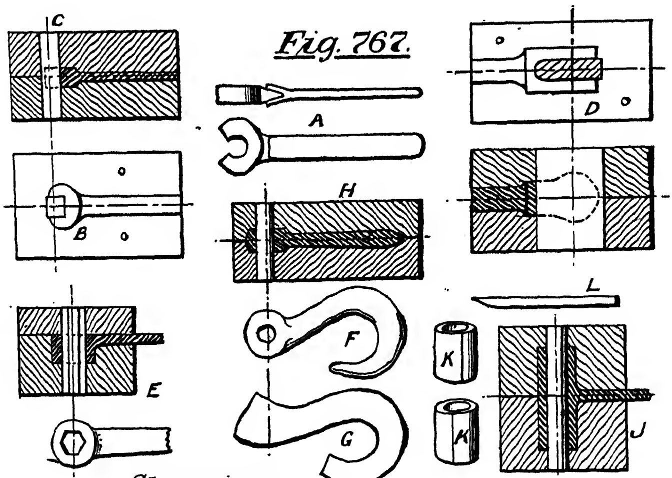
and Total Velocity $= \sqrt{2gH} + \sqrt{2\frac{Pg}{w}H} = 8\sqrt{H}\left(\sqrt{\frac{P}{w}} + 1\right)$

which value must be used instead of v on page 98.

Also $P = \frac{wv^2}{2gd} + w + P_s$ is more rigidly correct for mean total pressure, though the addition is only slight. Thus, by calculation a 5-cwt. hammer gives a blow of about 20 tons, half a ton of which is caused by P_s, the pressure of the steam. (*See* p. 98.)

P. 124. **Stamping.**—It is indeed remarkable how a difficult forging may be overcome by the use of top and bottom dies, and although the method has its limits, it may be pushed to very

Fig. 767.



Stamping.

extreme cases, if we partly forge by hand and partly stamp under a hammer. The metal should always be roughly forged or welded to shape, however, before placing in the dies, so as to dispose the fibre in the best direction for strength. Some seven examples are shewn: (1) The single-webbed crank, Fig. 121, needing no further explanation; (2) the centre for screwing stock,

C, Fig. 121*a*, which is first punched and roughed to shape, and then placed in suitable dies; (3) the spanner A, Fig. 121*a*, treated as in Fig. 767—that is, the jaw and shank are first scarfed and welded as at A, then placed in dies at B, and finally punched through at C before being removed; (4) the forked rod B, Fig. 121*a*, similarly rough-forged, and punched in dies as at D, Fig. 767; (5) the ring spanner E, Fig. 767, punched with a hexagonal punch; (6) the hook F, Fig. 767, first roughly bent as at G, and then stamped and punched as at H; and (7) the deep-eyed lever J, Fig. 767, first prepared by two rough rings K K, and a scarfed rod L, then placed in the dies and drifted as shewn. Whenever welding is done in the dies, the pieces must be raised to a very good welding heat; *see* B and J, Fig. 767.

P. 125. **Steelifying Iron.**—This process is of the same character as case-hardening, and is practised by making a powder having 1½ oz. of prussiate of potash, ½ oz. potassic nitrate, and ⅓ oz. sugar of lead; placing it upon red-hot iron, and reheating till the powder melts. Brightening a small portion of the iron, the colour is watched for as in tempering, and the quenching done in rain water. It is claimed that the hardening is very thorough, and makes the material suitable for cutting tools.

P. 128. **Hardening Steel.**—The hardness produced in cooling steel depends very much on the rapidity with which the heat is removed. Water is a good cooler, and is most used, but much harder results are obtained by cooling in mercury, and the hardest known by means of lead; the point of the tool, after heating, being pushed into a block of cold lead.

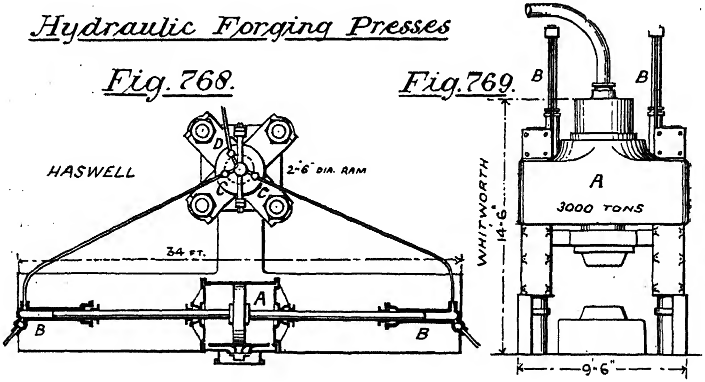
Hydraulic Forging.—Advocates of hydraulic pressure for heavy forging aver that steam hammers are done with, and that no more heavy hammers will be ordered. Yet hydraulic forging was used practically by Haswell in 1861, being proposed by Charles Fox in 1847, and many big hammers have since been built, proving that old prejudices die hard. At the same time it is fully conceded that the hammer blow merely compresses the exterior of the forging, and never satisfactorily reaches the centre, due evidently to the shortness of time occupied at each stroke. The advantage possessed by hydraulic forging would belong also

to any method that substituted a steady pressure for a blow, only that its rigidity makes water the most suitable medium. It is not, therefore, introduced on account of its storage qualities, and in fact the use of an accumulator or any possibility of a blow is disallowed at once. In most forging presses, then, the source of power—steam—lies near the press, and the water is merely a connection from there to the ram cylinder. The sole advantage of the system is that time is given for the metal to *flow* right through the forging thickness, and that this is no chimera, the statements of most celebrated engineers admit no doubt of.

Apparently hydraulic forging was suggested by Whitworth's compression of the fluid steel, and by the objection of his neighbours to the hammer noise; but it is doubtful whether the first practical press was due to Haswell, or to Gledhill, Whitworth's manager. Fig. 768 is a plan of Haswell's press, the steam piston

Hydraulic Forging Presses

Fig. 768. Fig. 769.

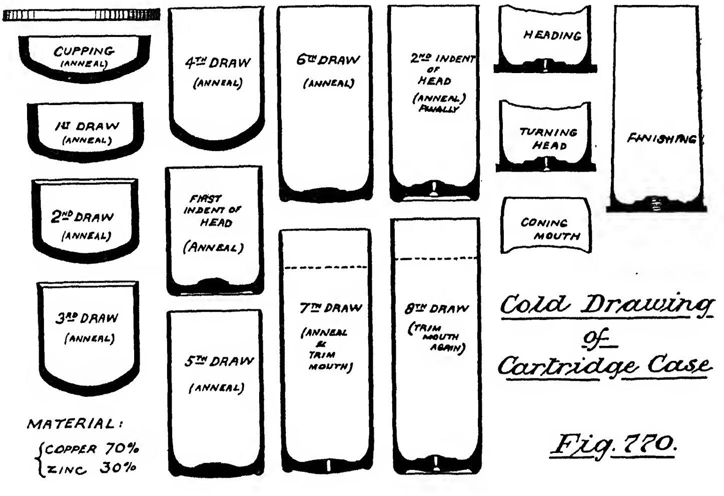


A being connected directly to the pumps; B B are non-return admission valves, and C C delivery valves, worked by powerful levers from an auxiliary steam cylinder. The piston travels a whole stroke in either direction alternately, valves C C being opened or closed as required, and the water is exhausted through a fifth valve, D. A smaller hydraulic ram placed above the main one serves to lift the latter by means of links. Whitworth's press is fed directly by steam-driven pumps, though the lifting rams are worked from an hydraulic accumulator, and his apparatus is easily

understood from Fig. 769, A being the main cylinder and B B the lifting rams.

Both methods have since been adopted satisfactorily, and two things have to be noted: (1) that immense rigidity of framing is required on account of the heavy pressures, 2 or 3 tons per square inch, and (2) that the difficulty of keeping valves and packings water-tight causes some makers to dispense with the former altogether. A very useful modern press, designed by the late Mr. Tweddell, is shewn at T, plate XV, facing p. 318, where the absence of pillars is a convenient arrangement.

Cold drawing of metals has long been practised in the manufacture of wire from more or less plastic materials, a hard



Cold Drawing of Cartridge Case

Fig. 770.

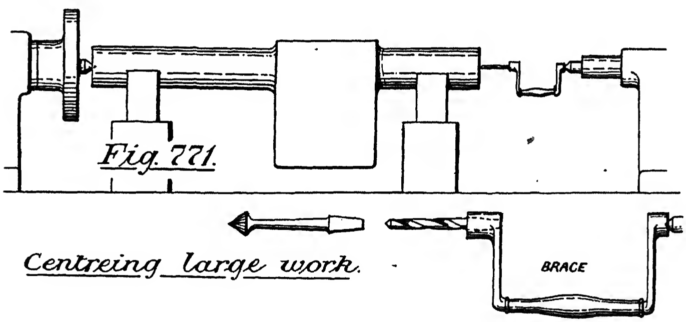
steel plate being drilled with a series of holes of gradually decreasing size, through which the material is passed in succession from largest to least. The principle is also applied to plates of the same materials, which are stamped by a regular series of dies until the required shape, often much removed from the original condition, is attained. Between every 'draw,' or nearly so, it is usual to anneal the work, for such forced flow of material produces

brittleness. The steel cylinders now used for storing compressed gases are thus made in one piece; so also are boiler tubes and cartridge cases. The last-mentioned have become of large size since the introduction of the quick-firing gun, and very heavy presses are therefore employed, the operations at Woolwich in drawing such a case for a 6 in. quick-firing gun being shewn in Fig. 770, as described by Sir William Anderson before the Institution of Mechanical Engineers in 1897.

Metal-spinning is a method of moulding thin flexible metal sheets upon wood blocks fixed in a lathe chuck, by means of a wooden tool or presser. In this manner knobs, teapots, and many other domestic articles can be built from spun hemispheres or saucers.

CHAPTER V.

P. 152. **Lathe Centres.**—The American practice is to use an angle of 60° for work up to 15 ins. diameter ($7\frac{1}{2}$ in. centres),

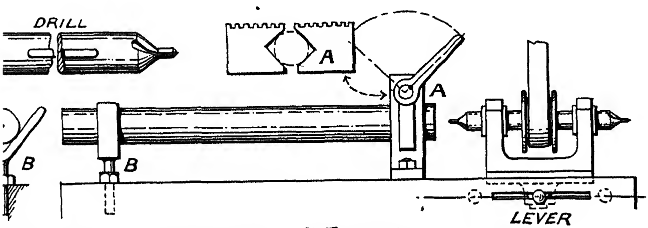


Centreing large work.

and 70° to 90° when above that diameter. Mr. W. H. Pretty, Wh. Sc., of Bedford, writes that an endeavour there to use 75° for small work (up to $\frac{3}{4}$ cwt.), and 80° for larger work, met with failure through changing of work, and a general standard of 80° being first adopted, was afterwards altered to 60.° to suit American tools. For work above $\frac{3}{4}$ cwt. he drills the ends with brace and bits, as in Fig. 771, by placing it on supports in line with lathe

s, and giving a feed by advancing the poppet screw; the drill being first used, and the countersink afterward.

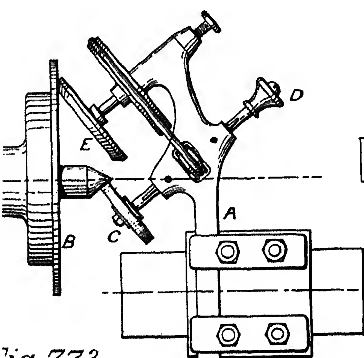
:ntreing Machine.—This is a simple but useful con-:e for rapidly preparing work for the lathe. The work is in a concentric chuck A, Fig. 772, and a V support B; the



:ig. 772. Centreing Machine.

being raised or lowered till the bar is level. The drill, ıg rapidly, is now advanced to the work by the lever, and ıuntersink and plain hole both drilled at one time by means : special drill-point shewn.

entre-grinding Wheel.—The common method of trueing he centres is to soften them by heating, turn them up, and



:ig. 773.

entre-grinding Wheel.

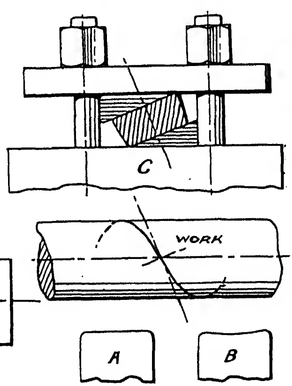


Fig. 774.

then re-harden. Fig. 773 shews a handy emery wheel for trueing up without preliminary softening. The shank A is fixed in the slide-rest so as to let a vulcanite wheel E roll upon disc B fastened to the catch plate, and the emery wheel C to just touch the centre point. The knob D, loose on the spindle, is now taken hold of to traverse wheel C, and the lathe mandrel is revolved, the connecting band between spindles E and C being a long helical spring.

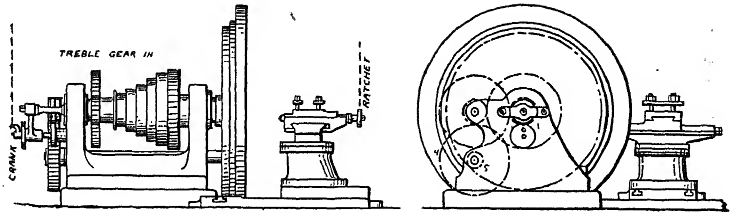
P. 157. **Water-finishing Tool.**—If a high and smooth finish is to be given to iron or steel, a broad, sharp tool is used, and plenty of water fed to its point while tooling. Quick speed and large feed are also supplied, and the tool-points are shewn in Fig. 774—A for a planer or shaper, and B for a lathe. The latter form can be understood by remembering that the relative path of tool to work is that of a screw, while the tipping of the tool at C permits its front to lie normally with the direction of travel.

P. 160. **Face Lathe.**—To give a clearer idea of this machine, a general drawing is provided in Fig. 775. The driving details have already been described for the break lathe, and the slide-rest needs no further description. The only point of difference lies in the bed, which, it will be seen, admits large diameters having small axial width. It is thus a *surfacing* machine.

P. 160. **Classification of Boring Machines.**—A short classification will give a better understanding of the many types of these machines. Thus, we may have:

1. Boring in the lathe: with moving work.
2. Special boring machine of lathe pattern.
3. Horizontal boring machine: with fixed work.
4. Vertical boring machine.
5. Snout-boring machine, for blind holes.

Lathe boring has been described at p. 160; but as universal tools are inadvisable, most machines being kept going with one class of work, it is better to construct a special machine (Class 2) of 'lathe-boring pattern, than to do much boring in the lathe



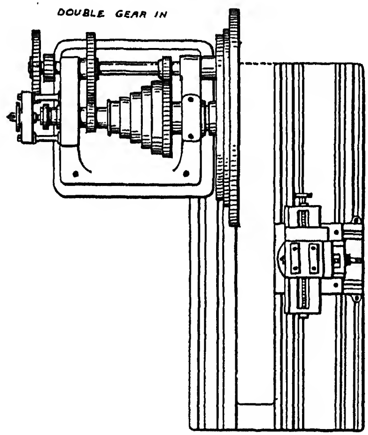


Fig. 775.

Face Lathe.

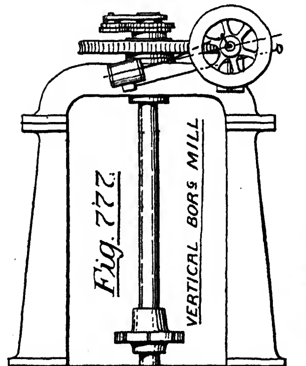


Fig. 777.

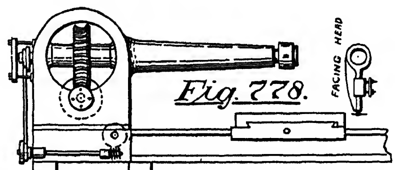


Fig. 778.

SNOUT BORG MACH.

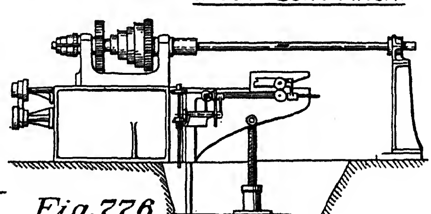


Fig. 776.

LATHE-PATTERN BORG MACH.

Boring Machines.

itself. Such a machine is useful and expeditious, if the work be not too heavy, and is illustrated in Fig. 776 by an example from Loudon Bros. of Johnstone, N.B. Some makers use two lifting screws when doing heavy work, and others support the boring bar on the saddle, as in Fig. 249, p. 237; but the main characteristics remain, such as deep bed and short length, and the movement of the work itself for feed.

Class 3, with fixed work, is well described on p. 161.

The fourth class, the vertical machine, has already been mentioned, and its advantages, truth of surface for large diameters, explained. The general design is shewn in Fig. 777, the construction being similar to those of Class 3, where the work is fixed and the tool fed along the bar by an epicyclic train of wheels at the upper end. Of course, the bar must be lifted vertically when removing the work, but there is less risk of accident than with the horizontal machine.

The snout-boring machine, Fig. 778, is made by Messrs. J. Buckton & Co., for cases where a bar cannot be passed through the work. Very large diameters cannot well be done, but most hydraulic cylinders can be conveniently bored. The driving is by worm gearing, and the feed is given to the saddle on which the work is bolted, lathe fashion. A facing head is also supplied.

P. 168. **The Slot-drilling Machine.**—Fig. 779 shews one

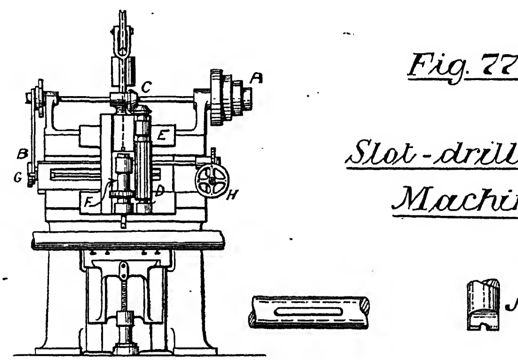


Fig. 779.

Slot-drilling Machine.

of these useful machines, to Messrs. Buckton's design. It is driven by speed cones A, from a countershaft in order to make

surface speed constant for various-sized drills. The horizontal feed to saddle E is given by the gear on the left at B; and the driving gear consists of mitre wheels C and a long Marlborough wheel D, thus permitting the tool to be set to any arranged depth. The slide F, carrying the tool spindle, may be adjusted to give the depth required, by means of spindle G, upon which is a small pinion gearing into a rack on the back of the slide. The table has the usual setting adjustments, and a hand feed is supplied at H. The form of drill is shewn at J, which first makes a hole of the proper depth, and is then traversed horizontally; but if the slot be very deep the work must be done in stages. (*See p.* 1019.)

P. 175. **Notes on Milling.**—When work is being fed to a milling cutter, the direction of motion of the work must be the reverse of that of the cutting tooth. If this precaution be not taken, the cutter will ride upon the work with great pressure, and its teeth be broken. Thus in the left view, Fig. 181, the work

Fig. 780

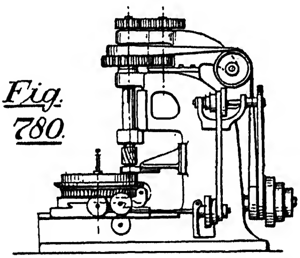
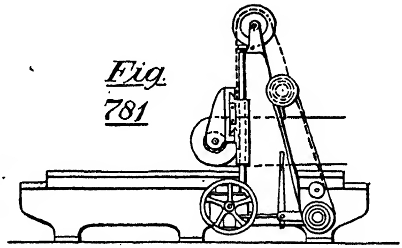


Fig. 781



Heavy Milling Machines.

should be fed from left to right. Further, when using a helical cutter, Fig. 181, the direction of helix must be such as to force the cutter on and not off the spindle.

Heavy Milling Machines.—Milling is being more and more adopted for repetition work. The mechanism of the Maxim gun, for instance, is all but automatically turned out, by milling machines of the pattern on Plate XII., intricate wavy sections being cut by *gangs* of mills, or several cutters strung on one spindle. A not less interesting development is that of the heavy

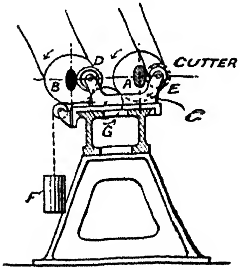
milling machine intended to directly supplant planers, shapers and slotters, even for ordinary unrepeated work. Over and over again has it been proved that this can be done with economy, despite renewal cost of mills, while the finish of the work is undoubtedly better. Fig. 780 is a vertical machine to do slotting-machine work, and the horizontal machine Fig. 781 similarly serves for planed work; the table movements giving feed in both cases, and not cut. (*See p.* 1020.)

Special Copying Machines.—The copying principle has an extreme illustration in such apparatus as the *copying lathe* and the *profiling machine.* The former was the invention of Blanchard, an American, and was used by him to make such articles as shoemakers' lasts, gun-stocks, &c. It is still much adopted for 'turning' the spokes and felloes of wooden wheels, and its principle will be understood from Fig. 782. There are two fixed headstocks, A and B, and two corresponding poppet heads. In B is placed a cast-iron copy, say a spoke, and in A a rough piece of wood. A sliding carriage C carries a roller D and a fly-cutter E of equal diameter, the latter being driven at high velocity by means of a belt. The roller D is pressed against the copy by the pull of weight F on the carriage, and the fly-cutter gouges out the wood in imitation of the copy. The feed must also be given. This is obtained by a *very slow* rotation of the mandrel B, which is communicated to the second mandrel through the idle wheel G, and as the roller D is moved backward or forward, the same movement occurs on the cutter E, so that the copy is accurately reproduced at any section, whatever its shape. At the same time a slow traverse is given to the carriage along the bed, thereby including all sections of the work.

The profiling machine is similar in character, but is arranged vertically, and is used for metal-cutting, its progenitor being retained for woodwork. Referring to Fig. 783, A is the copy, B the work, C the milling cutter, and D a 'dummy' to traverse the copy, the carriage E being pulled leftward as before. An enlarged view of the dummy at D_2 shews the cone shape which is required to gradually increase the depth of cut, by refixing at a higher position after each traverse. The bed is long, and similar to that of a planing machine, the feed being caused by a slow movement

of the table, as in milling machines. Flat connecting- and coupling-rods are good examples of the work done.

Fig. 782.



Copying Lathe.

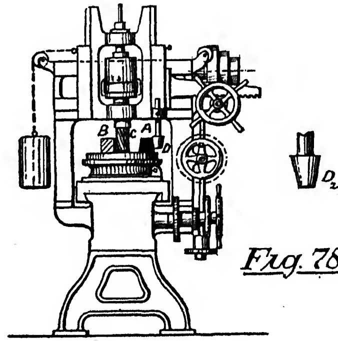


Fig. 783.

Profiling Machine.

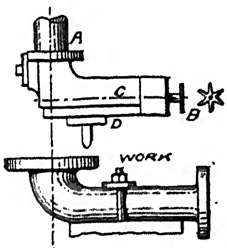


Fig. 785.

Facing Head.

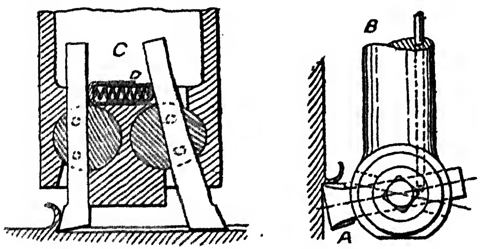


Fig. 784.

Reversible Tools.

Reversible Tools.—The endeavours of early manufacturers to increase the efficiency of reciprocating tools by making them

cut on both strokes have not been highly appreciated by users. Whitworth used a circular tool-box to his planing machines, and the tool-point was automatically moved through 180° after each stroke by cords and pulleys, much as the plate-planer tool is now moved (see C, Fig. 285, p. 295). Lack of rigidity caused the abandonment of this tool for good machines, however. Two recent double-acting tools, by Messrs. J. Buckton & Co., are shewn in Fig. 784. The tool A is suited to a slotting machine, and is automatically reversed by rod B, on which are tappets. The main difficulty here, and whenever *one* tool has two edges, is the difficulty of sharpening symmetrically. The planing tool-box C is an improvement in this respect, for both tools can easily be set at their proper heights. The tools are fixed in slots made in discs, so that when one tool is at work the other trails on the return, the spring D keeping the acting tool to its cut. Both arrangements are said to work very well in practice.

Facing Head.—Small articles such as pipe flanges are very economically surfaced in a drilling machine by adopting a facing head as in Fig. 785. It is fixed to the spindle by a coned shank and collar as usual, and a radial feed is given to the tool-box D by the star B and screw C, the former striking a fixed projection at every revolution of the spindle.

CHAPTER VI.

P. 200. **Turret-head Lathe with vertical Mandrel.**—The special tool shewn in Fig. 785*a* is called by its designers, the Richards Machine Tool Co., a 'Universal Turning Machine,' but is apparently better described as above. It may be understood by a reference to the description on p. 200. The mandrel is supported on a footstep, and driven by worm gear, and there are two vertical slides corresponding to a lathe saddle and slide rest. The turret is shewn provided with tools for turning piston rings from a cylindrical casting, for which purpose *a* turns the top, *b* roughs the thickness which *c* finishes, and *d* parts to correct width. For the last operation the tools are gradually fed through the work by turning the hand wheel *e*. (*See p.* 978.)

P. 212. **Originating a Surface Plate.** — Mr. W. H.

Pretty, Wh. Sc., finds it possible to save much time and labour when originating surface plates or straight-edges, by giving the workman a tabulated statement of the order of procedure, so arranged as to systematically reduce the errors of manipulation. The plates having been stamped with numbers, (1), (2), (3), in conspicuous places, and the planing-tool marks eliminated with a smooth file, he follows this order:

(*a*) Using (1) as a standard: bed (2) to (1); bed (3) to (1); then bed (2) to (3), working equally on each.

(*b*) Using (2) as a standard: bed (1) to (2); (3) is already bedded to (2); then bed (3) to (1), working equally on each.

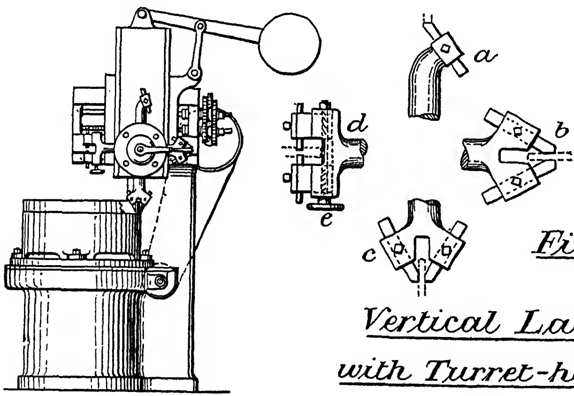


Fig. 785a.

Vertical Lathe, with Turret-head.

(*c*) Using (3) as a standard: bed (2) to (3); (1) is already bedded to (3); then bed (1) to (2), working equally on each.

(*d*) Using (1) as a standard: bed (3) to (1); (2) is already bedded to (1); then bed (2) to (3), working equally on each.

This cycle of operations is repeated until sufficient accuracy is attained. When straight-edges are being trued up they should be occasionally reversed end for end, so as to eliminate all possible errors.

P. 214. **Screw-cutting.** — There are no fewer than five

ways of cutting a screw-thread upon a spindle, which may be thus enumerated:

(1.) Cutting with *stock and dies* as described at p. 192. This principle is defective, for reasons there mentioned. The corresponding socket is screwed with *taps*.

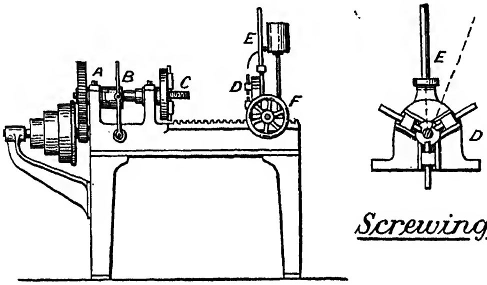


Fig. 786.

Screwing Machine.

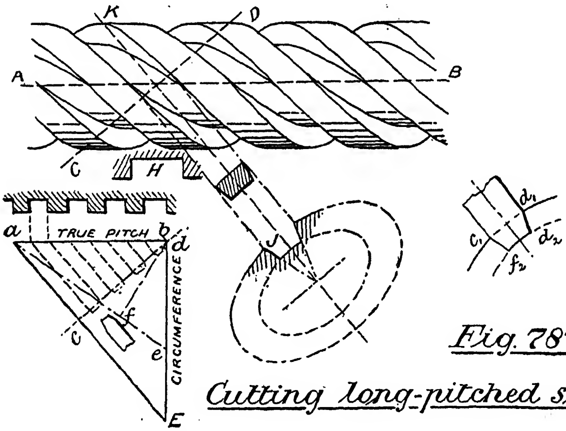


Fig. 787.

Cutting long-pitched screws.

(2.) Screwing in the *lathe*, described at pp. 147, 212, and 484. This is the only method giving a perfect screw, except that next mentioned, and is really equivalent to scale copying.

(3.) *Copying* is generally performed in a turret-head lathe as at p. 200, but is also shewn at 47, Fig. 317, p. 349, for screwing stay tubes. The copy is often of a larger scale than the work.

(4.) By *Chasing* only. On p. 212 the chaser is described as

a finishing tool, when cutting screws in the lathe. It is, however, sometimes used both to start and finish the thread, being held by the hand throughout the operations. It is easily seen that such a method has precisely the same objections as stock and dies, but even to a worse extent, for 'drunken' threads, of varying pitch, are often produced by inexperienced workmen. This is because the tool has very short directing surface, even though that surface may have the correct angle.

(5.) By *Screwing Machine.*—This method is the same in principle as stock and dies, and is merely a quicker and more automatic way of doing the same work. In Fig. 786 the general form of the machine is that of a short lathe with fixed headstock A, a clutch B for putting the mandrel in and out of gear quickly, and a powerful driving gear. The work is held in a concentric chuck C, and the screwing dies are in themselves a sort of concentric chuck, operated by the lever E. The mandrel being revolved, the cutting head D is brought over the work by the pinion and handwheel F until the dies are in position to start: the lever E is then pulled leftward by the advancing cutters, being partly helped by the workman's hand at E. The operation may be repeated until the stud is of correct diameter, as indicated by the angle of the lever E. With plenty of lubrication the thread may be cut during one traverse, and when the dies are blunted they can be recut upon a master tap placed in chuck C.

Cutting Long-pitched Screws. — When the thread angle becomes 45° or over, it may be questionable policy to cut it in a lathe. Remembering that a screw is formed by an axial traverse and a rotation, either motion may be called the *cut*, the other being the *feed*; but the latter should always be preferably the slower or smaller motion. Thus, in the machine for rifling guns, the traverse being large and the rotation small, the tool is propelled axially, the holder being a long bar with the tool secreted in the end, in the manner known as the 'tiger's claw,' from the fact that it is sheathed on entering, and automatically shot out on the outward or cutting stroke. During withdrawal the tool bar is rotated by a rack and pinion, the amount of rotation being fixed by the inclination of a bar along which the rack arm travels. In such manner any long-pitched screw may be cut.

But whatever machine may be adopted, there are some general rules that apply to every case, though more particularly if the angle be great or the pitch large, for then any deviation from rule is more apparent. Imagine a screw A B, having, say, four threads wrapped round its elementary cylinder. The combination of cut and feed will cause the work to travel under the tool in the direction K J, and the tool front must therefore be set normal to this line. The section at H will not shew the real shape or width of tool, but that at J, which is taken across the line C D. The comparison of the two may be shewn by a diagram : thus, if *a b* be the true pitch, over four threads, measured axially, and *d* E the outside circumference, *c d* will be the pitch measured normal to the threads. By setting out the threads on *a b*, and projecting them on *c d*, the true width of tool at thread top may be found,

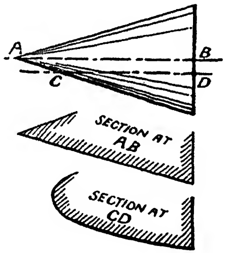


Fig. 788.

Incorrect Taper-turning.

as at $c_1 d_1$. Again, by making *d e* equal to circumference at thread bottom, we have *f d* as the normal pitch upon which the threads are to be again projected from *a b*, giving f_2 d_2, the width of tool at thread bottom. Finally, the curvature of tool point will be that due to the ellipse obtained as section on line C D. The same rules apply to broad traversing tools for plain lathe-work, so far as curvature of tool and angle of path is concerned.

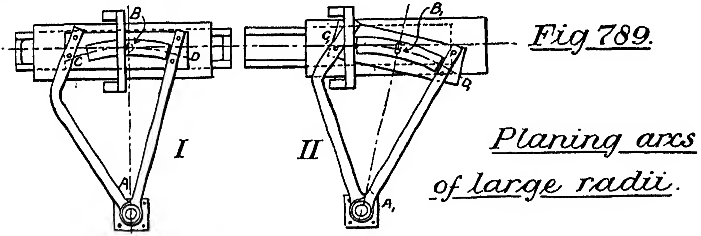
P. 226. **Incorrect Taper-turning.** — When cylindrical work only is being turned, the *correctness* of the cylinder is not in the least marred by the height at which the tool is set, though the beauty of the surface may be very much affected. But the turning of a tapered surface or cone requires special care in this respect, and it is of the highest importance that

the height of the tool point should be exactly level with the centre line of the work; for the section A B of a cone, Fig. 788, through the axis, is a triangle, for which a straight-line feed would be suitable, but the section at C D, below the axis, is a hyperbola, and a straight-line feed would simply produce a reversed hyperbola instead of a true cone.

P. 249. **Tooling Circular Arcs.**—The methods of tooling arcs of large radius may thus be classified:—

1. Milling or slotting, with hand feed.
2. Milling in a profiling machine having a curved copy.
3. Special milling apparatus, p. 752, using a property of the circle.
4. Planing on a pivoted table controlled by a rod equal in length to arc radius.
5. Turning on a large face plate in a vertical lathe, whose short mandrel is sunk in the ground.

Numbers 1, 2, and 3 have already been described. No. 5 is practised at Woolwich on racers or roller paths of large radius. But as large arcs are not often required, it is better to fit up a planing machine in the manner shewn at Fig. 789, where two



positions are given. Taking position I., let a line A B be drawn immediately under the tool point, and let a stud B be fixed to the table. Next, let a triangular frame A C D, one side of which is the small table C D, be pivoted on stud B, and further pinned at A; the hole in C D being slotted to permit the planing table to travel. The work to be planed is now fastened to C D, and the planing commences; then, by referring to position II., it will be found that the curve C D, struck from A, will always lie under the tool point whatever the table position.

P. 256. **Turning Balls.**—Brass and gun-metal balls are much used for feed-pump- and safety-valves on account of absence of sticking which they ensure. To turn them in the lathe, a cup chuck in hard wood—A, Fig. 790—is provided, into

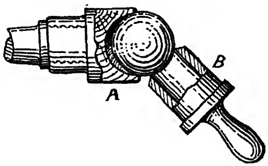


Fig. 790.

Ball turning.

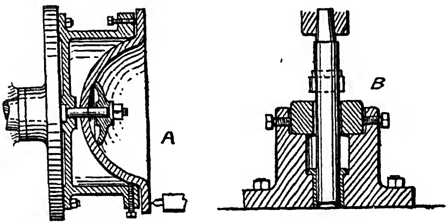


Fig. 791.

Jigs.

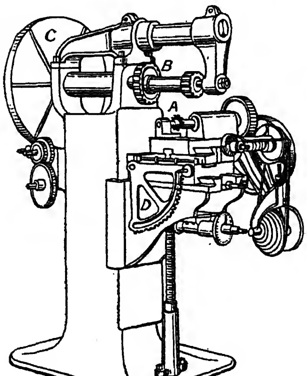


Fig. 792.

Automatic Gear-cutting Machine.

which the ball is fixed. This form of chuck permits the work to be constantly changed in position; and, if this be carefully done, it will be clear that any section will be a circle, which is

the only requirement in a true sphere. After the ball is made as perfect as possible with the usual turning tools, it is finished with the tool B, made from steel tube, which is rocked to and fro as the lathe revolves, the ball being often removed.

P. 274. **Jigs.**—These appliances may be constructed so as to hold work for other operations besides drilling bolt holes, and Fig. 791 shews two arrangements—A being a special chuck for holding a dome cover while turning, and B a vice for supporting a loose collar for boring.

Cutting Wheel Teeth.—The formation of the teeth of wheels, either on paper or in the workshop, has been mentioned at various places in this book as follows:—

Kind of Tooth.	Patterns.	Moulds.	Cutting in Metal.	Describing Teeth Curves.
Spur	p. 60	p. 31	pp. 175 & 180	pp. 510 & 517
Bevel	p. 60	p. 31	pp. 256 & 753	p. 519
Worm	p. 58	p. 10	p. 274	—

A few more words on this very important subject will not be out of place.

Spur-wheel Teeth.—A milling cutter is always used to remove the interspaces, but the 'blank' to be cut may be mounted in one of various ways. On p. 180, the dividing heads are used for support, but this method is only suitable for small diameters. When wheels of large diameter form the regular work of a shop, special machines are adopted; in many cases so constructed as to automatically rotate the wheel through the pitch arc after every cut. Such a machine is shewn in Fig. 792, elaborate but effective, where A is the cutter, B the work, C the dividing wheel, and D a sector for tilting the table to suit bevel gears.

Yet another method, shewn in Swasey's machine, Fig. 793, is based on the fact that if a number of wheels be cut so as to each gear with a given rack, they will all gear the one with the other.

The 'rack' is represented by the gang of cutters A, which simultaneously rotate and travel slowly in an axial direction. While cutting takes place, however, the blank is also rotated on its own axis, by means of change wheels, in such wise as to accurately roll upon the 'rack,' with no slip whatever. The consequence is, that the spaces cut out are not quite of the same shape as the cutter teeth, but are so widened that a real rack of the cutter section would roll perfectly with the wheel thus cut. The axial traverse of the cutters is given by the fixed cam C, which gradually

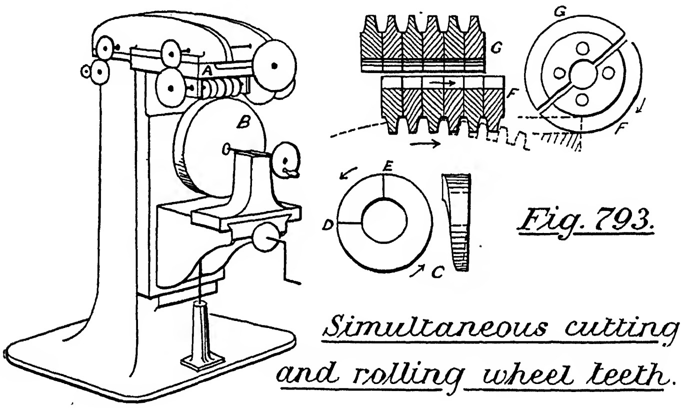


Fig. 793.

Simultaneous cutting and rolling wheel teeth.

thickens from D to E, through about three-quarters of a turn, and returns rapidly between E and D. Now the cutters are in halves, and while one set F is cutting, and *advancing* axially, the other set G, being out of the cut, is returning. The method appears complicated, but in reality is very simple, and the wheels thus cut will gear together most correctly, without backlash. By a more recent method, a single emery wheel, of correct section, rotates in one place while the wheel is really rolled along a line parallel to the grinding axis; and thus cast wheels may be trued up.

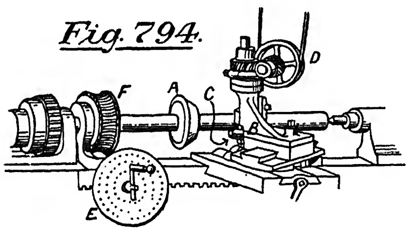
If a milling machine be not at hand, very good wheel-cutting can be done in the lathe. The lathe centres support a mandrel which carries the blank, and the milling cutter is fixed on a

vertical spindle which revolves in a bracket set upon the slide-rest, being there driven from a counter-shaft pulley overhead. The whole apparatus is seen in Fig. 794, where A is the blank, B the bracket, containing the cutter spindle C driven by the belt D, and E a dividing plate which gives the correct pitch turn through the worm wheel F.

Bevel-wheel Teeth.—It has already been explained at p. 753, that true bevel teeth can only be cut with a conical feed, and that milling cutters are useless except for approximations, or for roughing out before using an exact tool. (*See Appendix V.*, p. 986.)

Worm-wheel Teeth.—These also are cut by many methods

Fig. 794.



Wheel-cutting in the Lathe.

which more or less approach accuracy. The simplest way, though by no means a good one, is to rough out the spaces with a milling cutter, as on p. 58, and then to apply the worm, as there shewn, but using the file to trim down the thick portions of the teeth where marked by red ochre from the worm.

The second method is to use a *hob* (*see* p. 274). This tool is obtained by turning a steel worm of proper size and shape, cutting out milling teeth upon it, backing these off at top and sides for clearance angle, and finally hardening. Such a tool is highly expensive to make, and there is considerable risk when hardening; and unless finished with an emery wheel is likely to be untrue. Very few sizes can therefore be afforded, and wheels must be kept to standard pitch. Again, supposing the hob provided, there are still two methods of applying it. The spaces are always roughed out with a cutter, in order to save the hob, and the wheel being mounted freely on a stud, the hob is rotated while

being gradually brought into gear with the wheel. Usually the wheel is not turned round automatically, and then the hob has to do the double duty of rotating the wheel and cutting the teeth. The fault of this method is precisely that of stocks and dies in screw-cutting, for the worm thread has a different angle at the top

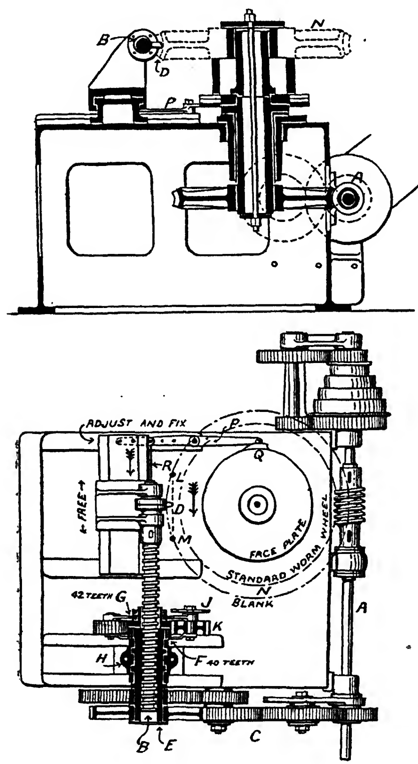


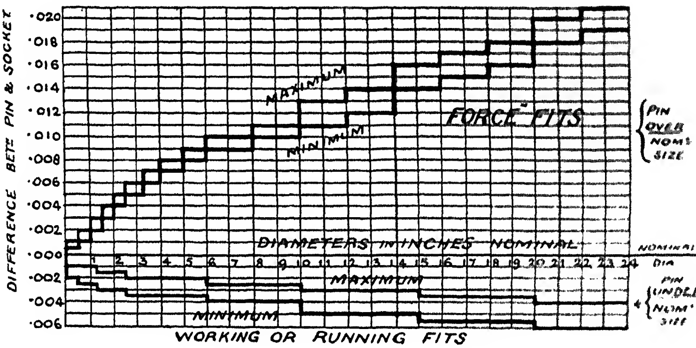
Fig. 795. Cutting and forming Worm-wheel teeth simultaneously. (Mr J. H. Gibson's Machine.)

to what it has at the bottom. Better results are obtained by using a machine like a short lathe, and mounting the wheel on a vertical shaft passing through a fixed saddle, while the hob is driven between the lathe centres. The wheel shaft is then rotated at the proper velocity regarding the worm, by means of change wheels.

The machine in Fig 795 was described by Mr. J. H. Gibson, in a paper read before the North-East Coast Institution in 1897. A is the driving shaft, connected to the cutter shaft B by change wheels, arranged to give the actual relative motion between the worm and wheel. The cutter D makes a purely circular rotation, which, however, traces a helical cut on the wheel blank N. The shaft B is driven through nut E, wheel F, and wheel G, the latter riding on a feather key in the screw; but while cutting, all these pieces rotate solidly in the bearing H. Only *once* per revolution of the wheel blank N, the lever P is pressed out by the cam Q, bringing rod R forward, as shewn by arrow, so that the star wheel J may catch R and cause wheel K to roll round G and F. As G has 42 teeth and F 40, these two wheels thus move relatively to one another, and the screw is shifted axially forward by the worm pitch. The blank being fixed on the face plate, and the lever P set just past the cam, the cutter is placed at L, and the machine started. The result of the various motions is to make a series of light cuts or scratches on the blank, marking out the interspaces; but, when N has turned round once, the lever P moves, and, as previously explained, the cutter takes up a new position on the helix of the imaginary worm. The cutting still proceeding automatically, the *same* grooves are simply cut a little deeper, and so the cycle of operations is repeated till the cutter emerges at M, by which time every interspace will have been completely finished; for the cutter, representing the worm, will have been presented in every one of its many positions, and the wheel will have been truly 'rolled.' Finally the worm is turned, with the cutter as template, and of a pitch equal to screw B, of which various sizes are kept.

✓ *P. 277.* **Standard Fits.**—Mr. Arthur G. Fuller has recently made very careful experiments to determine the proper *working fit* clearance, which should vary with the size of the object. As regards *driving fits*, the pin must be slightly larger than the hole, but the amount depends on the strength of drive required. *Force fits* require a still larger pin, but the size would again depend on whether the force applied were that of a lever, screw, or hydraulic press. The last-mentioned fits Mr. Fuller estimated from the average of all the experience he could collect,

and the results of his investigations are set out in Fig. 796 for working and force fits. Drive fits are to have from $\frac{1}{4}$ to $\frac{1}{2}$ the largeness given for force fits, the former for light drives, and the latter for heavy ones. The high and low gauges should be so



Standard Fits. *Fig. 796.*

NB. DRIVING FITS ONE-HALF FORCE FITS.

made that the clearance (or largeness) should always be between the maximum and minimum amounts shewn, and the method is known as the 'limit' system. (For shrink fits *see* p. 842.)

CHAPTER VII.

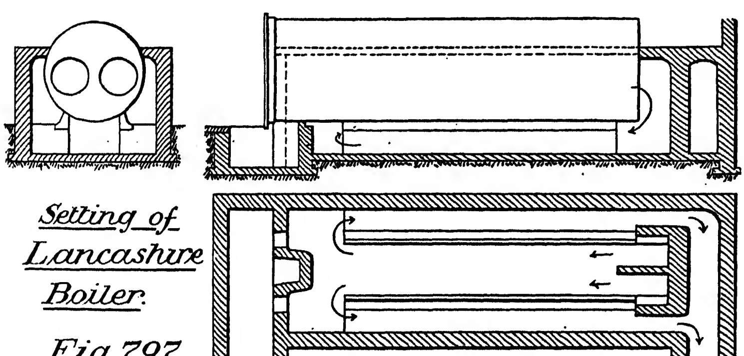
P. 286. **Caulking,** if moderate, is not objectionable, but *split* caulking is as bad as it can be, and should be carefully guarded against. It consists of splitting off and then turning in at the joint a strip of iron about $\frac{1}{32}$ in. wide, and although it may stop a leak at the time, ultimately breaks out worse than ever.

P. 313. **Increased Pressure in Rivetter when closing.**—The pressure of water in a rivetter being 1500 lbs. per square in. at first, is increased by about 50% at the moment of closing the rivet, or to about 2250 lbs. per square in.; the

cause being the sudden arrest of the accumulator weight and consequent absorption of inertia.

P. 330. **Electric Welding.**—There are four systems at present in use, the Thompson, Benardos, Zerener, and Voltex. The first is useful for wires or bars, heat being caused by their resistance, while the Benardos uses the arc, and is suitable for repairs generally, but both require large installations. The Zerener process avoids passing the current through the material, the arc between two carbons placed in line being deflected upon the work by means of electro-magnets. The Voltex system also adopts two carbons; but they are set mutually at about 45°, and magnets are unnecessary. Both the last-named systems are self-contained, being carried about with ease, and it is claimed that there is a saving in current of 30 % in the Voltex over the Benardos process. Also that the double carbon apparatus does not harden the parts so as to prevent machining, as in the other cases. The Voltex requires a current of 80 volts, with 120 ampères for welding, with 30 ampères for brazing, and with 5 ampères for soldering. (*See p.* 1055.)

P. 332. **Setting for a Lancashire Boiler.**—This type of boiler not being complete without external flues, which also



Setting of Lancashire Boiler.

Fig. 797.

act as boiler supports, sections have been shewn in Fig. 797. The direction of draught is indicated by arrows: through the

furnace tubes, underneath the boiler, and returning along the side flues to the chimney. Where the brickwork touches the boiler it should be as narrow as possible to avoid corrosion or any unseen wasting, and should there be luted with fireclay instead of mortar. The whole setting is lined with firebrick and inclines slightly towards the front in order to drain.

P. 337. **The Field Boiler.**—This is principally interesting on account of the Field tube, which has been much applied where quick steaming has been required. The tube consists of an outer or blind member, A, Fig. 798, having water inside and

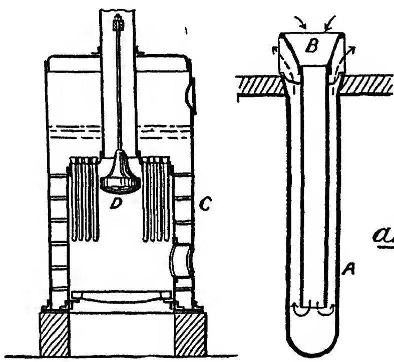


Fig. 798.

Field Tube and Field Boiler.

furnace gas outside, and the inner tube B is supported by feathers at the top, its funnel mouth entrapping the downward current, thus causing the upward current to ascend by tube A, and promoting the disengagement of steam. The boiler C shews the tubes in position, hung from the firebox crown, D being a baffle plate to retard and reflect the draught.

P. 339. **Water-tube Boilers** having been greatly favoured for certain purposes during recent years, a more extended account seems advisable. These boilers were tried in France as early as 1871, and have been used in the French Navy since 1874. In the meantime the Babcock-Wilcox boiler was introduced in various parts of the world. The necessity for a boiler for the English Navy suited to high pressures, combining light weight

with safety, and having rapid steam-raising properties, created the Thornycroft and Yarrow boilers, at least for the smaller boats; after which the Belleville boiler (originally introduced for French boats in 1879) was naturalised in England for the larger ships. Meanwhile, the simultaneous development of the cylindrical or 'Scotch' marine boiler proceeded, and now it is a question which is to hold permanent ground. The consensus of opinion seems to be in favour of retaining the Scotch boiler for passenger and cargo service, but especially for the latter, while the special advantages of the water-tube boiler fit it for war vessels. On land, the high efficiency of the Lancashire boiler and its steady steam-supplying properties make it a difficult rival: neither is it necessary to save space or weight on land. Taking now the water-tube boiler alone, we may class its advantages:

Advantages of Water-tube Boilers.

1. Will give higher pressure steam than cylindrical boilers.
2. Repairs, though probably more frequent, are easily made.
3. Steam can be raised more quickly from cold water, there being little risk of deformation.
4. Circulation is *systematic*, rising by the tubes, and returning by 'downcomer.' Circulation, however, does not increase rate of evaporation, but the reverse: its function is to keep steam moving and avoid dry plates.
5. Lighter than other boilers for same power, because *heating* surface is largely *containing* surface as well, whereas a separate (and heavy) envelope is required for the latter purpose in other boilers. Also heating surface is lighter per square foot than in many other boilers.
6. Less space for same power.
7. Safer at high pressures. This, however, is dependent on good circulation and on a fairly small steam reservoir.
8. Portability: can be carried about in sections and fitted up afterwards.

The disadvantages of this boiler seem to be its doubtful economy in regular use, and the fact that only smokeless coal can be burned. Its greatest advantage is its favouring high

pressures, which soon reach their limit in cylindrical boilers; so, stated briefly, a less economical boiler is adopted to secure a more economical engine. It is a mistake to suppose all water-tube boilers fit for forcing: the following classifies the principal types regarding this quality:—

LIST OF TYPICAL WATER-TUBE BOILERS.

For Natural Draught only.

Babcock-Wilcox.	Oriolle.
Root.	Niclausse.
Lagrafel d'Allest.	De Naeger.

Capable of some Forcing.

Belleville.	Herreschof.

For Forced Draught.

Thornycroft.	Normand.
Yarrow.	Du Temple.

The four last-mentioned are most suitable for fast-speed war

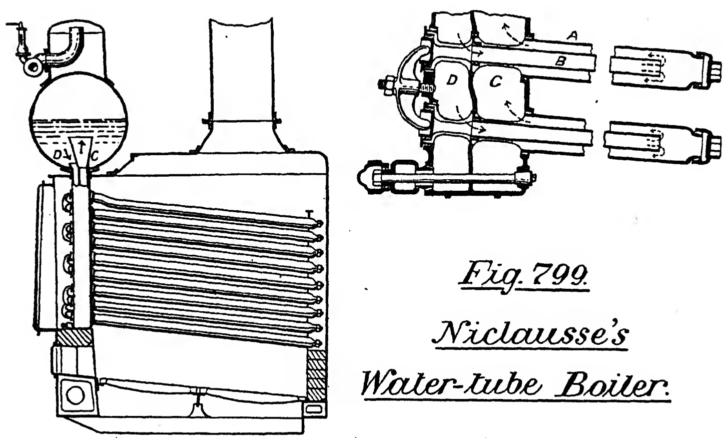


Fig. 799.
Niclausse's Water-tube Boiler.

vessels, such as 'destroyers,' which require increased combustion on occasion.

The *Babcock-Wilcox* boiler has been already described, and may be taken as a type of natural-draught boiler. The *Niclausse*

deserves mention on account of its use of an inclined Field tube, the inner portion of which serves as downcomer; and there is only one header, a double one, placed at the front. In Fig. 799, A is the rising tube and B the downcomer, the rising header C entering the steam receiver at a higher point than the falling header D. The tubes are easily removed from the front.

The *Belleville* boiler, now considerably adopted on English battleships, is shewn in Fig. 800. A is a non-conducting casing,

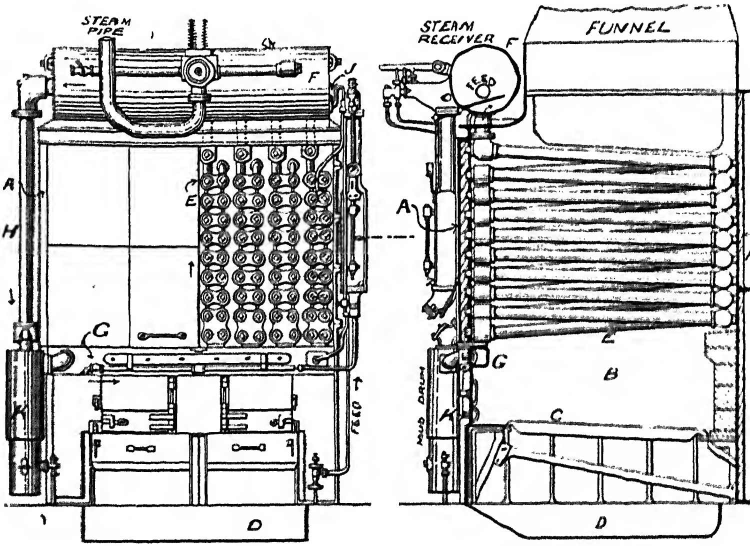


Fig. 800. The Belleville Boiler.

lined with firebrick at B, and containing firebars C, ashpit D, and tubes E. The tubes are divided into eight sets or 'elements,' each of which is a *coil* having its upper end connected to the steam receiver F, and its lower end to the feed collector G, which is a continuation of the downcomer H. The feed water, automatically controlled by a float-governed check valve, enters the receiver F at J, and is distributed along the lower portion, flowing then through pipes H and G, from the latter of which it enters

the element coils, rises, and passes into the receiver, beneath the baffle plate which separates the hot steam from the colder feed. The circulation is therefore caused by difference of density. K is a mud collector, or blow-off chamber. It is usual to make steam at a much higher pressure than is required, and let it pass through a reducing valve on its way to the cylinder; this gives drier steam by throttling, ensures a more perfect deposition of lime and magnesia, gives a steadier pressure at the engine, and permits the use of a smaller boiler for a given power. (*See p.* 919.)

In the *Thornycroft* boiler, Fig. 801, there is one steam receiver

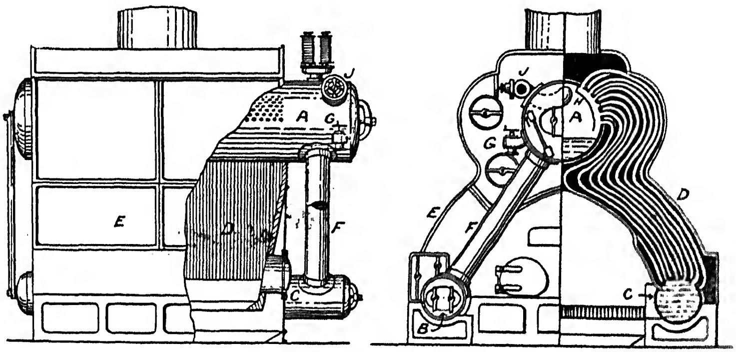


Fig. 801. The Thornycroft Boiler.

A, but two mud drums B and C, and circulation is due to a rise in the tubes D D and a fall in the downcomers F, outside the casing. The feed enters at G, the water being delivered along the receiver bottom, while two principal features are the entry of the water tubes into the steam space of the receiver, and their tortuous form for the purpose of meeting the draught normally. H is a baffle plate, and J the steam pipe.

The *Yarrow* and *Normand* boilers are similar to the Thornycroft, but the first has straight tubes, while those of the second are but slightly bent; in both cases they enter the steam drum at the bottom. The *Du Temple* is of the same type. The *Root*, *Lagrafel d'Allest*, *Oriolle*, and *De Naeger* are similar to

the Babcock-Wilcox, and the *Herreschof* is a coil boiler with pump circulation. (*See pp.* 918, 993, *and* 1061.)

Fusible Plugs (see also p. 755). Fig. 802 is a section of

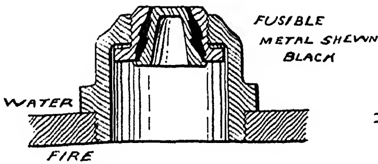


Fig. 802.

Fusible Plug.

one of these as introduced by the National Boiler Insurance Company. The white-metal should be renewed annually.

CHAPTER VIII.

P. 367. **Modulus of Resilience.**—The greatest elastic stress is often called *proof stress.* Then, from diagram 2, Fig. 326, we may define resilience as 'half proof load × proof strain,' or

$$\text{Resilience} = \tfrac{1}{2}\,W\Delta$$

$$\text{But } \Delta = \frac{fl}{E} \qquad \text{and } W = fa$$

$$\therefore \text{ Resilience} = \tfrac{1}{2} fa \frac{fl}{E} = \frac{f^2}{E}\ (\tfrac{1}{2} \text{ volume}).$$

The quantity $f^2 \div E$ is termed the *modulus of resilience*, and measures resistance to impact per cubic inch of the bar.

Example 61.—Compare the resilience per unit volume of two bolts A and B, in which for $\frac{1}{10}$ its length A has a sectional area ·8 of the remaining $\frac{9}{10}$: and B has for $\frac{9}{10}$ its length a sectional area ·8 of the remaining $\frac{1}{10}$ (Hons. Mach. Const. Exam. 1891).

In each case let f = unit stress on smaller area
and $\cdot 8f$ = unit stress on larger area.

Also let a = larger area, and $\cdot 8a$ = smaller area.

Then, taking bolt A, total resilience

$$= \left(\frac{f^2}{E} \times \frac{\cdot 8a \times 1}{2}\right) + \left(\frac{(\cdot 8f)^2}{E} \times \frac{a \times 9}{2}\right)$$

Resilience per unit volume $= \frac{3{\cdot}28 f^2 a}{E} \div$ vol. $= {\cdot}334 \frac{f^2}{E}$

Similarly, taking bolt B, total resilience

$$= \left(\frac{f^2}{E} \times \frac{{\cdot}8a \times 9}{2}\right) + \left(\frac{({\cdot}8f)^2}{E} \times \frac{a \times 1}{2}\right)$$

Resilience per unit volume $= \frac{39{\cdot}2 f^2 a}{E} \div$ vol. $= {\cdot}465 \frac{f^2}{E}$

$$\therefore \frac{\text{Resilience per unit vol. A}}{\text{Resilience per unit vol. B}} = \frac{334}{465} = \underline{{\cdot}714 : 1}$$

And bolt B is more economical for resisting shock, shewing why it is better to turn down a bolt shank as at p. 402 to the diameter at bottom of screw thread.

P. 369. **Testing Machines.**—The *Drop-testing Machine* has always been favoured by railway companies for proving rails and car axles. The apparatus is simple, consisting merely of a large anvil on which are supports for the beam to be tested, a pair of upright guides, and a falling weight that can be raised to any definite height within them. The French railways use a 'monkey' (falling weight) of 440 lbs. with a drop of 11 ft. 6 ins., the rail supports being 3 ft. 7¼ ins. span, and the anvil weighing 10 tons. Messrs. Cammell use a weight of 1 ton, a fall of 20 or 30 ft., and a rail span of 3 ft. Rigidity and inertia of anvil are of considerable importance, but the chief difficulty is to attain constant rigidity. The Pennsylvania railroad has therefore placed its anvil, weighing 17,500 lbs., upon 12 stiff helical springs, each having two coils of 8″ and 5½″ diameter respectively, the outer one of $1\frac{3}{16}$″ steel and the inner of $\frac{13}{16}$″ steel, 9⅛″ long uncompressed and 5½″ compressed. These can support 80,000 lbs., and exert a constant resistance whatever the condition of ground. The monkey weighs 1640 lbs., its maximum fall being 43 feet, and the supports are 3 ft. apart for axle-testing. It is specified that the axles shall not deflect more than 5½″ with the first blow, delivered from a height of 23½ ft., and shall stand five blows before fracture.

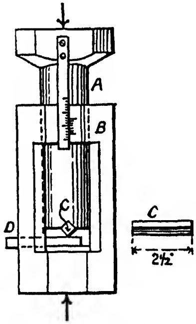
Testing for Hardness.—Hardness may be defined as the resistance to permanent deformation, often a property of great importance, and a means of testing hardness with accuracy is very much to be desired. All hardness tests depend upon making an indentation in the material by means of a harder substance, but the

measurements are variously made. Mr. Thos. Turner loads a diamond point till it just scratches, and the load measures the hardness, while others have used points, knife edges or punches of hard tool steel, and have severally measured volume, depth, or length of the print. Prof. Unwin's method is more scientific, for although he uses only a knife edge of tool steel, he has, by plotting his experimental results, deduced the following formula :

$$\text{Relative hardness} = \frac{p^n}{i}$$

Where i = depth of indentation in inches.
p = pressure per inch width of knife edge.
n = 1·2.

His apparatus, Fig. 803, consists of a plunger A sliding in a



Prof. Unwin's apparatus for testing hardness

Fig. 803.

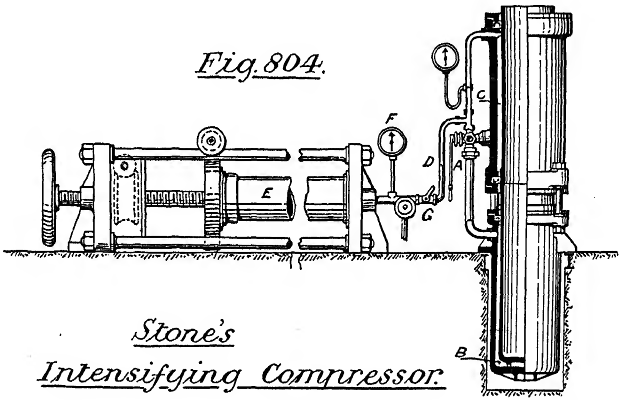
socket B, and pressing on the piece of excessively hard tool steel C placed on the specimen D. The whole is loaded in a testing machine, as shewn by arrows, and both load and depth of impression measured, the former being gradually increased, and time allowed at each step. In no case must the specimen be stretched.

TABLE OF RELATIVE HARDNESSES.
(Measured by Prof. Unwin.)

Cast Steel	554
Brass	233
Mild Steel	143
Aluminium (cast)	103
Copper (annealed)	62
Zinc (cast)	41
Lead (cast)	4

P. 375. **Intensifying Compressor.**—The differential hydraulic principle shewn in Fig. 727, p. 737, is a convenient means of obtaining high pressure with small load, if little quantity of water be desired, or a double-acting apparatus be permissible for continuing the stroke. It may equally well be applied to testing machines for intensifying town's water, and is so shewn in Fig. 804, which represents Messrs. J. Stone & Co.'s apparatus for

Fig. 804.



Stone's Intensifying Compressor.

testing pipe mains. The town's water is admitted through cock A, and acts at B on the larger area, thus intensifying the water in C, where the load is received on the annulus. The pressure water then passes along pipe D to the main E being tested, and the pressure is shewn at F. The cock A is so arranged that town's water may be first admitted into C, D, and E, after which the flow is diverted to B and intensification takes place, the ram rising till its stroke is complete. If the test be not then finished, cock G is closed, while C and B are placed in equilibrium by means of cock A, and the ram is once more lowered. Again closing A, the pressure is intensified instantaneously in C, and G being opened, the test continues.

In a double-acting apparatus made by the same firm, a pair of rams are used, one being up while the other is down, and the cock, A, one to each ram, is opened and closed automatically by

tappets, so that the testing is continued without attention until stopped by hand.

P. 381. **Test Specimens.** — Professor Carpenter has experimented on specimens of various lengths, 2″, 4″, 6″, and 8″, from which he deduces that the ultimate strength (as shewn by highest point of stress-strain curve) is independent of specimen length, and that percentage extension (ductility) at maximum load is pretty uniform. The ductility at rupture is, however, variable, being inversely as length of specimen up to 8″ long, and afterwards constant. This indicates the need of a standard length of not less than 8″; but if maximum load and extension only are required, length is of no importance.

P. 385. **Stress-Strain Diagrams.**

Mechanical Hysteresis.—If tension experiments are made upon

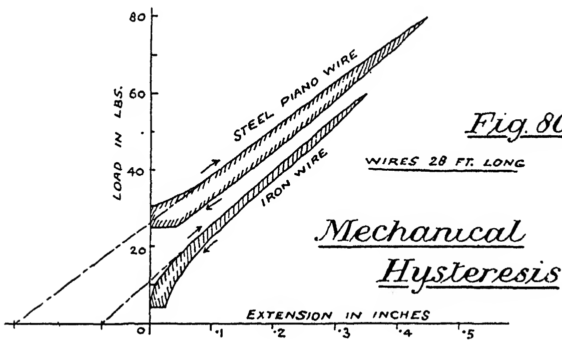


Fig. 805

rods or beams well within the elastic limit, by means of loads gradually increased from zero to a maximum and then gradually decreased to zero again, the *ascending* stress-strain curve will not agree with the descending one. The first will follow the true elastic line, but the second will slope more steeply and meet the strain base at a point somewhat to the right of the origin. The apparent permanent strain, called *lag*, will gradually disappear, however. The area enclosed by the two curves has been called *mechanical hysteresis* (from its similarity to magnetic hysteresis)

and represents a loss which is probably due to heating. Fig. 805 shews experiments on wires 28 feet long, where hysteresis is very clear. The iron wire had a diameter of ·049″, and the steel wire ·045″. (*See p.* 1071.)

Influence of Temperature on Strength.—This may be separately considered for low and for high temperatures.

At low temperatures Professor Rudeloff tried seven different materials:

1. Rivet iron.
2. Rolled iron.
3. Hammered iron.
4. Acid open-hearth steel.
5. Basic Bessemer steel.
6. Spring steel.
7. Crucible steel.

And the temperatures for every material were three, 64°, – 4°,

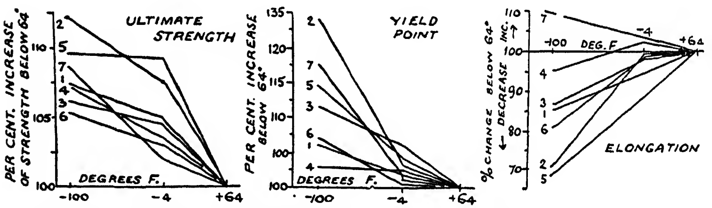


Fig. 806. Influence of temperature on strength.

and – 100° Fahr. Measurements were made of ultimate tensile strength, yield-point strength, and percentage elongation in $3\frac{1}{2}$ ins., the results appearing in Fig. 806, where each specimen is numbered. Much, apparently, depends on the chemical composition of the material, but generally the ultimate strength is raised rapidly at first and slowly afterward, the yield point slowly at first and rapidly afterward, while percentage elongation is generally decreased. The material is therefore less capable of resisting shock at low temperatures. Professor Dewar finds a strength increase of 50 to 100% at – 295° Fahr.

At high temperatures metals decrease both in strength and ductility. Copper alloys become much weaker, but cast iron is little affected. Probably Professor Unwin has given most

attention to strengths at high temperatures, and he states that iron and steel usually gain slightly in strength up to 500°, but after 600° decrease rapidly. For the copper alloys he deduces the maximum strength:

$$f = a - b\,(t^\circ - 60)^2$$

where the values of a and b are as follows:

	a.	b.
Copper	14·8	·000014
Gun-metal (cast)	12·5	·000050
Phosphor Bronze (cast)	16·1	·000026
Brass (cast)	12·5	·000024
Brass (rolled)	24·1	·000028
Delta Metal (rolled)	31·3	·000041
Muntz Metal (rolled)	24·7	·000029

Experiments are made by surrounding the specimen with a box of oil heated by gas-jets, the measuring apparatus being outside. (*See p.* 1074.)

EXPERIMENT ON WHITWORTH COMPRESSED STEEL.
BY PROFESSOR GOODMAN.

	At 60° Fahr.	At 400° Fahr.
Maximum stress per sq. in. ...	34·03	31·83
Elastic limit per sq. in.... ...	21·97	20·45
Extension in 10 ins.	16%	11·7%

P. 390. **Wöhler's Law.**—Unwin's formula, p. 390, being empirical, cannot be proved from first principles, so its application will be further shewn in the next problem.

Example 62.—Two steel bars, having a static breaking load of 30 tons per square inch, are stressed in tension, the one from 5 to 6 tons and the other from 1 to 6 tons per square inch. Find the actual breaking strengths to be assumed for the respective methods of loading.

Now stress variation must be put in terms of maximum stress, and this again in terms of f_2, the new breaking stress, thus :

$$S = \frac{\text{highest stress} - \text{lowest stress}}{\text{highest stress}} \times f_2$$

Case I. Ultimate static stress = 30 tons

$$S = \frac{6-5}{6} f_2 = \tfrac{1}{6} f_2$$

$$\therefore f_2 = \tfrac{1}{12} f_2 + \sqrt{30^2 - (2 \times \tfrac{1}{6} f_2 \times 30)}$$

Solving the quadratic, $\underline{f_2 = 27{\cdot}25 \text{ tons square inch.}}$

Case II. Ultimate static stress = 30 tons

$$S = \frac{6-1}{6} f_2 = \tfrac{5}{6} f_2$$

$$\therefore f_2 = \tfrac{5}{12} f_2 + \sqrt{30^2 - (2 \times \tfrac{5}{6} f_2 \times 30)}$$

Solving as before, $\underline{f_2 = 25{\cdot}7 \text{ tons square inch.}}$

The value of x in this formula may be 1·5 for iron and 2 for steel in general, but varies considerably, the deductions from Wöhler's and Bauschinger's experiments being :

Iron	x	*Steel*	x
Bar Iron (Wöhler) ...	1·33	Axle Steel (Wöhler) ...	1·83
Bar Iron (Bauschinger) .	1·67	Axle Steel (Bauschinger)	1·91
Bar Iron (ditto) ...	1·53	Rail Steel (ditto) ...	2·00
Plate Iron (ditto) ...	1·60	Spring Steel (Wöhler) ...	2·14
		Boiler Steel (Bauschinger)	1·53

P. 393. **Average Stresses.**—The following additional figures are here given, all in tons and inches :

Material.	Tensile Stress		Extension per cent. at breaking.
	Breaking	Safe	
Manganese Steel (14% Mn) ...	60	10	44
Nickel Steel (3% Ni for plates) ...	37	7½	20
Nickel Steel (3.35% Ni for forging)	45	8	27½
Silicon Bronze	28	5	30
Aluminium Bronze	30	5	25
Delta Metal (forged)	30	5	—
Delta Metal (cast)	20	3½	—
Sterro Metal	27	4½	—

(*See p.* 1075.)

P. 400. **Thick Cylinders.**—It appears from Lamé's formula (p. 399) that in an originally unstrained cylinder, if the pressure from inside be greater than f the tensile strength of the material, no amount of thickness can prevent bursting. This difficulty can be to some extent overcome by the principle of initial stressing, p. 758. It has also been found that if the fluid pressure in cast-iron cylinders be gradually increased beyond the calculated limit, the *internal* diameter may be permanently stretched, and then p has been known to reach 3 tons sq. in. with safety.

The formulæ used in designing built-up guns, deduced from Lamé, is here given (see also Fig. 749).

p_0 = internal pressure on firing
p_1 = pressure between A and B tubes on firing
p_2 = pressure between B and C tubes on firing.

t_0 = maximum hoop tension in cylinder A
t_1 = maximum hoop tension in cylinder B
t_2 = maximum hoop tension in cylinder C.

r_0 = internal radius of cylinder A
r_1 = internal radius of cylinder B
r_2 = internal radius of cylinder C
r_3 = *external* radius of cylinder C.

Then, $$p_0 = \frac{r_1^2 - r_0^2}{r_1^2 + r_0^2}(t_0 + p_1) + p_1 \qquad p_1 = \frac{r_2^2 - r_1^2}{r_2^2 + r_1^2}(t_1 + p_2) + p_2$$

$$p_2 = \frac{r_3^2 - r_2^2}{r_3^2 + r_2^2} \cdot t_2$$

To obtain these results, each outer tube must be smaller than the next inner tube by an amount called *shrinkage.*

$$S_2 = \left\{ \begin{matrix} \text{Shrinkage between} \\ \text{A and B} \end{matrix} \right\} = \frac{p_0 - p_1 + t_1 - t_0}{E} \cdot 2r_1$$

$$S_3 = \left\{ \begin{matrix} \text{Shrinkage between} \\ \text{B and C} \end{matrix} \right\} = \left\{ \frac{t_2 + p_0 - (p_0 - p_2)\left(\dfrac{r_2^2 + r_0^2}{r_2^2 - r_0^2}\right)}{E} \right\} 2r_2$$

The formula for p may be extended to four or reduced to two

tubes, for it is seen how p_0 follows from p_1, and so on, but for shrinkage a further value is given for clearness, thus:

$$S_4 = \left\{ \begin{matrix} \text{Shrinkage between} \\ \text{C and D} \end{matrix} \right\} = \left\{ \frac{t_3 + p_0 + (p_0 - p_3)\left(\frac{r_3^2 + r_0^2}{r_3^2 - r_0^2}\right)}{E} \right\} 2r_3$$

The radii are first assumed, varying approximately in geometrical progression outwardly. A limit is next placed on the hoop tensions, and p_2 found. From p_2 we pass to p_1 and thence p_0 (the safe gas pressure) is deduced. Finally, the shrinkages are calculated to give these pressures. A rough shrinkage rule is given on p. 400.

P. 400. **Shrink Fits.**—There are three methods of fitting a cylinder rigidly in a socket, viz. by *driving*, *forcing*, and *shrinking*. In all three, the cylinder must be larger than the socket by an amount determined by experience. The 'largeness' for driving and force fits is given on p. 826; and for shrink fits, where the outer portion is heated before slipping over the cylinder, the following simple rules may be adopted:—

$$S = {\cdot}0025 \times \text{diameter of hole}$$

if the parts are very thick and unresisting; but

$$S = {\cdot}0035 \times \text{diameter of hole}$$

if thinner and more elastic. Care must be taken not to heat higher than is absolutely necessary, and to prevent endlong sliding by means of clamps. (*See p.* 825.)

P. 402. **Strength of Bolts.**—Engineers have disagreed considerably as to the stress which comes on fluid-tight covers, though the problem is easily determined. First, suppose the flange and seating be quite rigid, and no packing be placed between the surfaces; also that the necessary tightness is obtained by an initial 'screwing-up' stress in the bolt, as at A, Fig. 807. If now the fluid stress be exerted, the flange will tend to take the condition B, but as it is evident that (the flanges being rigid) the surfaces cannot separate till the pressure exceeds the initial stress, it follows that the bolt cannot stretch till such stress is exceeded; and the bolt stress must either be that due to screwing or to

pressure, whichever be greater, but not both. As, however, an excess of fluid pressure would cause the joint to leak, the initial stress cannot be exceeded.

Secondly, suppose an elastic packing be placed between the surfaces, and let us examine the problem by the elementary apparatus at C, Fig. 807. Between two walls *d* and *e* a light block *f* is supported, by a thin wire spring *b* and a strong india-rubber bar *a*; and let the walls be separated so as to produce a tensile stress, say, of 10 lbs. in each spring. Further, imagine

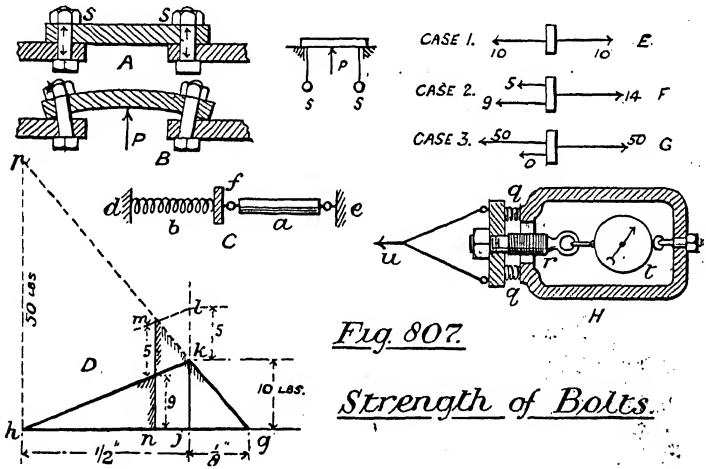


Fig. 807.

Strength of Bolts.

that the wire spring is thus caused to extend $\frac{1}{2}''$, while the rubber bar stretches $\frac{1}{8}''$. The stress-strain diagram is shewn at D, where $jk = 10$ lbs., $hj = \frac{1}{2}''$, and $jg = \frac{1}{8}''$. Now let a force of 5 lbs. be put upon the block *f* so as to pull it leftwards, thus increasing the stress in *a* and decreasing it in *b*. Shewing this on the diagram, produce gk and jk: make $kl = 5$ lbs., draw $lm \parallel kh$, and $mn \parallel lj$. The shaded portion shews the new diagrams, indicating the stress in *b* as 9 lbs., and that in *a* as 14 lbs. If now the force of 5 lbs. be gradually increased, the stress in *b* will decrease; and when a force of 50 lbs. is reached as at hp, the spring is entirely freed from stress. The three cases are shewn by the static diagrams E, F, and G.

We next apply the elementary case to the model H. The spring *b* becomes the packing *q q*, the rubber bar the bolt *r*; and the two models C and H are therefore under similar conditions, only that *q q* are in compression instead of tension. Applying the same numbers, the screwing stress is 10 lbs., felt equally on *r* and *q*, and shewn on dial *t*, and a force of 5 lbs. being exerted at *u*. there is a tensile stress of 14 lbs. in *r*, while the compressive stress in *q* is reduced to 9 lbs. Thus, any real conditions may be ascertained by a diagram such as D, if the resiliences of bolt and packing be known. If the packing be practically rigid, we approach the case A, but the flanges always have a certain elasticity.

In practice, the real difficulty is to find what stress the workman will cause in screwing up; hence the rules on p. 402 are usually adopted, where the screwing stress is made a ratio of the pressure, and the latter taken as the only guide in calculation. Also the pressed area is measured to the inner edge of the bolts.

P. 422. **Shaft Couplings.**—Mr. Archibald Sharp's coupling, Fig. 808, is a combination of box and flange, and is an undoubted

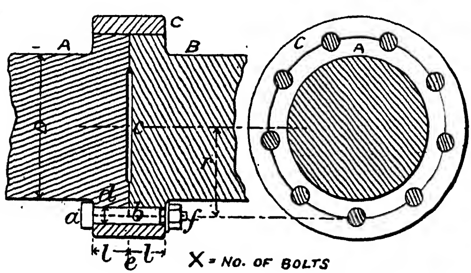


Fig. 808.

Sharp's Coupling.

improvement on the latter as regards strength distribution. The bolts receive shear stress along planes shewn by dotted lines, and the twisting effort in A is transmitted through surfaces *a b*, *b c*. Similarly, B receives the twist through surfaces *f b*, *b c* The usual flange-coupling bolt has a shear stress over the whole cross section, as *e b c*, but here ring C binds the outer halves of every bolt, passing the strength *a b* to *b f*, and the bolts are only

weakened at the half cross-section *b c*. Thus only about half the usual bolt strength is required.

Equating the strengths of shaft and bolts, and neglecting the small half cross-sectional bolt strength at *b*,

$$f_s \frac{\pi D^3}{16} = (\tfrac{3}{4} f_s \, X l d) r$$

$$\therefore \quad d = \cdot 262 \frac{D^3}{X l r}$$

The number of bolts may be $X = 2 \cdot 5 \sqrt{D}$

the nearest whole number being taken. If also the ring c be made large enough to cover the bolt washers, it will have ample strength.

P. 426. **Deflection of Helical Springs.**—In his difficult and laborious investigations on the strength of square and rectangular shafts, St. Venant found that the greatest stress and strain occurred at the middle of the (longest) side of the sections. Now in formula for θ, p. 424, d is evidently the diameter of greatest stress; therefore, for square wire,

$$f_s = \frac{wr}{\cdot 208 s^3} \qquad \text{and } \Delta = \frac{2 f_s l r}{C s} = \frac{w n r^3}{C d^4} \, 60.5$$

while for rectangular wire (*see p.* 421),

$$f_s = \frac{wr}{\cdot 2944 \left(\frac{b^2 h^2}{\sqrt{b^2 + h^2}} \right)} \qquad \text{and } \Delta = \frac{2 f_s l r}{C b} = \frac{w n r^3}{C \left(\frac{b^3 h^2}{\sqrt{b^2 + h^2}} \right)} \, 42.6$$

P. 430–2. **Moment of Resistance, Moment of Inertia or Second Moment, and Centres of Gravity.**—As area has no mass, purists now object to the term 'Moment of Inertia of Area,' substituting the more reasonable '*Second Moment*;' and '*Centroid*' is similarly adopted instead of 'Centre of Gravity of Area.' Also we may define the n^{th} *moment* of an area round a given line, as that obtained by dividing the area into very small pieces, and multiplying every piece by the n^{th} power of its distance from the given line; or, algebraically,

$$n^{th} \text{ moment} = \Sigma (a h^n)$$

We thus have 1st, 2nd, and 3rd moments of an area, each of which has its use, and all easily found by a graphic construction. Referring to Fig. 809, let the given area PP_1, be imagined to

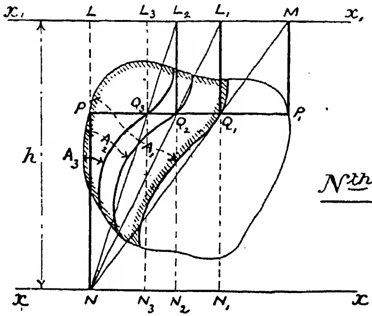


Fig. 809.

N^{th} Moment of Area.

rotate round axis $x\,x$. Draw $PP_1 \parallel x\,x$, and take any new line, x_1x_1 at any distance h, and $\parallel x\,x$. Project P_1M and PN, and join MN, giving a point Q_1. Do this for several horizontal intercepts, and obtain the shaded 1st moment area PQ_1 In like manner project Q_1L_1 and join L_1N, giving the shaded 2nd moment area PQ_2. Similarly the 3rd moment area is obtained from area PQ_2. Then, calling the areas respectively A_1, A_2, and A_3,

$$\left.\begin{aligned} \text{1st moment} &= A_1\,h \\ \text{2nd moment} &= A_2\,h^2 \\ \text{3rd moment} &= A_3\,h^3 \end{aligned}\right\} \text{of the original area round } x\,x,$$

and any higher-powered moment can be obtained by continued process.

Proof.—Let PP_1 be an element of area having width e.

$$n^{th} \text{ moment} = \Sigma\{PP \times e \times (PN)^n\}$$

$$\text{Now } \frac{LM}{PQ_1} = \frac{h}{PN} \quad \text{and } LM = PP_1$$

$$\therefore\ PP_1 \times PN = h \times PQ_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (a)$$

$$\text{and 1st moment} = \Sigma\{PP_1 \times e \times (PN)^1\}$$

$$= \Sigma(PP_1 \times PN \times e) = h \times \Sigma(e \times PQ_1) = \underline{A_1 h.}$$

$$\text{Again } \frac{LL_1}{PQ_2} = \frac{h}{PN} \quad \text{and } LL_1 = PQ_1$$

$$\therefore \quad PQ_1 \times PN = h \times PQ_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (b)$$

and 2nd moment $= \Sigma\{PP_1 \times e \times (PN)^2\}$

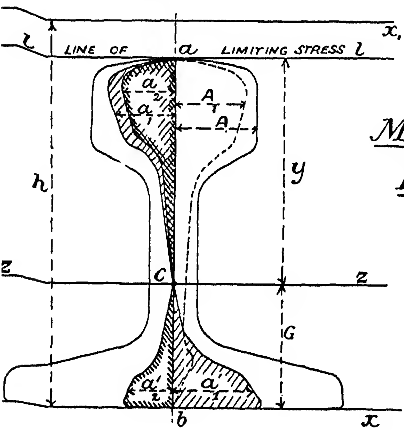
$= \Sigma\,(e \times h \times PQ_1 \times PN)$ ... ... by substitution from (a)

$= \Sigma\,(e \times h^2 \times PQ_2)$ ... ... by substitution from (b)

$$= h^2 \times \Sigma\,(e \times PQ_2) = \underline{A_2 h^2}$$

igher moments may be similarly proved.

pplying the construction to beam sections, we must first find entroid. Imagine any given section $a\,b$ (Fig. 810). Find its



Moment of Resistance.

Fig. 810.

ıoment A_1, round xx say, using any height h. Dealing only he right-hand half for simplicity, we have from the definition ıtre-of-gravity,

$$A_1 h = A\,G \qquad \text{Hence, } G = \frac{A_1 h}{A}.$$

gives the height of neutral axis $z\,z$ by a much more simple ccurate method than those on p. 432, especially if a plani- is obtainable. We next require I, the 2nd moment, round ıing preferably the reference distance y to line of limiting in constructing the curves. Treating the left side, to avoid ıion, every horizontal strip is referred to $l\,l$, and its projection to C, *producing if necessary*, till the original strip is crossed; hus the areas a_1 a'_1 are found, on *opposite* sides of the ıl centre line. Continuing the process on areas a_1 a'_1, the

2nd moment areas $a_2\ a'_2$ are obtained on the *same* sides of the vertical. Now the real value of I will be found by doubling our results, for we have only used half of the section; hence,

$$I = 2(a_2 + a'_2)y^2 \qquad \text{and} \quad Z = \frac{I}{y} = 2(a_2 + a'_2)y$$

or generally,

$$\underline{\text{Moment of Resistance}} = \underline{f\,(\text{2nd moment area})\,y}$$

if the reference distance y has been adopted. In cast-iron beams y_t is always taken, which is less than y_c (see p. 435), and the resulting curves are slightly changed, but the construction explained must still be rigidly followed. It will be seen that this method is superior to that at p. 430, though the latter is still left in the text as sometimes convenient. (*See p.* 1079.)

P. 441. **Fixed Beams.**—When beams have their ends fixed as in Figs. 399 and 400, the B_m curve may be found graphically, by supposing it the algebraic sum of two moment curves, one caused by the 'action' of the free load, and the other by the 'reaction' (a couple) in the wall itself. These opposing curves must cover exactly equal areas: for, considering the upper fibres of the beam say, the total effect of the load is to shorten them; but, remembering that the total length of the beam is unalterable on account of its fixedness, the reactionary couple must entirely eliminate the aforesaid compression. The sum of the strains is therefore zero, and because $B_m \propto f \propto \delta$ in uniform sectioned beams, the sum of the B_m (average $B_m \times l$) due to *free* load must equal the sum of B_m due to wall couple.

Taking the special cases of Figs. 399 and 400, the first is easily solved, but the second or uniformly loaded beam will be further explained by Fig. 811. Now the B_m curve for a free beam is a parabola, where $bd = Wl \div 8$; and the wall moment is $efgh$, where $eg = \frac{2}{3}(bd)$, to make areas A and B equal. The final B_m curve is shewn shaded at C. Lastly, to find contraflexure points, take any distance x between k and l.

$$B_m \text{ at } x = \frac{Wl}{12} - D = \frac{Wl}{12} - \left(\frac{W}{2}x - \frac{x}{l}W \cdot \frac{x}{2}\right).$$

$$= \frac{W(l^2 - 6xl + 6x^2)}{12\,l}$$

Let x be increased till it equals kl. Then B_m = zero, or

$$l^2 - 6xl + 6x^2 = 0$$

from which, solving the quadratic,

$$x \text{ or } kl = \cdot 211 l.$$

Let a fixed beam A B be loaded in any general or unsymmetrical manner as in Fig. 812. The curve of B_m for free load is found at $a\,b\,c$. The reactionary moment curve $d\,e\,f\,g$ is next to be obtained. The two areas $a\,b\,c$ and $d\,e\,f\,g$ must now not only

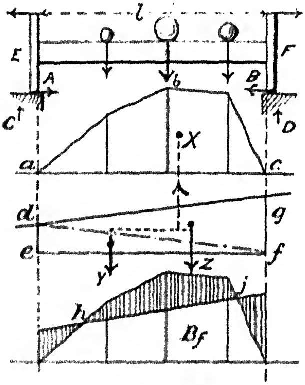


Fig. 812

Fixed Beams.

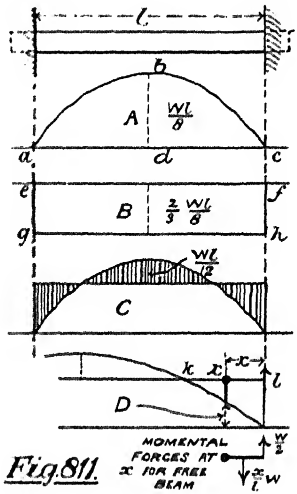


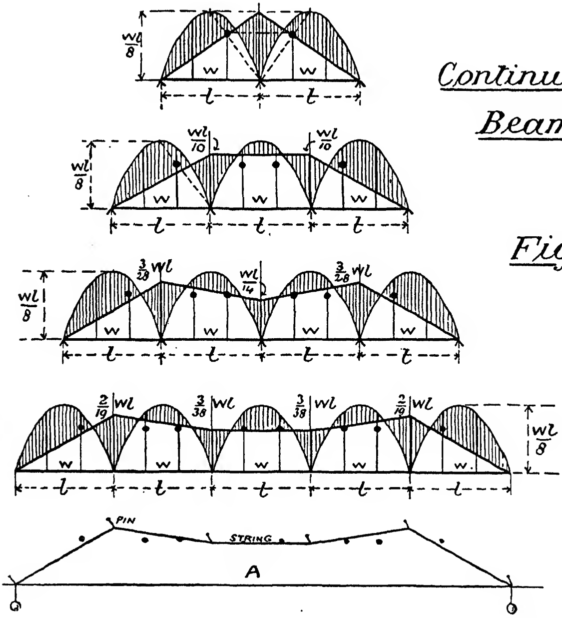
Fig. 811.

be equal, but their centroids must lie on a common vertical. If then X shew the value of the free area $a\,b\,c$ as also the position of its centroid, Y and Z must be placed at $\frac{1}{3}l$ and of such value as to balance X in the manner of parallel forces, where Y + Z = X. Then area $d\,e\,f$ = Y and area $d\,g\,f$ = Z, from which the moments $d\,e$ and $g\,f$ can be deduced. Lastly the resulting moment curve B_f is obtained by superposition.

Case VIII., Fig. 401, p. 443, is shewn in double at the top part of Fig. 813 next page.

P. 445. **Bending Moments for Continuous Girders.** —The previous methods may be further extended to these cases

Four examples are given in Fig. 813, where for two spans we merely have a duplicate of Fig. 401, p. 443. In all cases the parabolas $W\,l \div 8$ are first drawn for *free* beams, and these are next opposed by a curve of reactionary moments, consisting of straight lines shewing zero at the extreme ends. In the figure the actual moments are drawn and stated, as found from Clapeyron's formula, the shaded curve giving the final B_m; but the second or



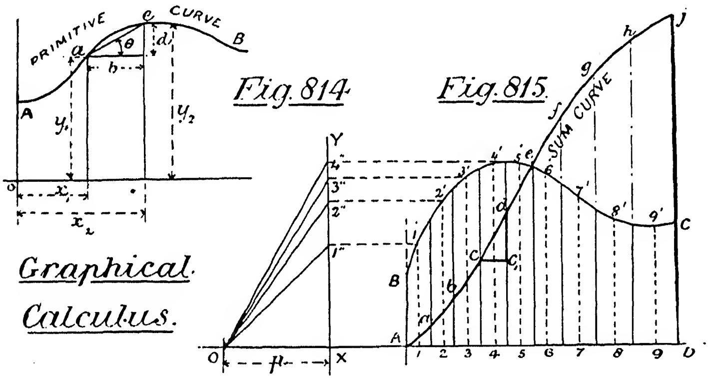
Continuous Beams

Fig. 813.

opposing curve may be very closely obtained in a graphic manner. Divide every span in thirds, drawing vertical lines, and on every vertical except the outside ones put a mark at $\frac{2}{3}$ of $Wl \div 8$. Then pass the second curve through these points so as to give the best average results, which is easiest done by string, pins, and weights, as at A. This method, which is due to Claxton Fidler, is clearly an extension of the principles already explained, and has the advantage of being applicable to any mode of fairly uniform loading, however much it may vary from span to span. (*See p.* 953.)

Graphical Calculus.—Let a curve be drawn with given ordinates and abscissæ, and be called a *primitive curve.* A second curve may be constructed to the same base, which will shew the gradually increasing value of the area under the first curve, as we travel say from left to right; and this second curve we shall call the *sum curve.* It represents the integration or summation of the first curve. A third curve can next be drawn shewing the rate of rise or fall of the primitive curve, and this we shall term the rate- or *slope-curve,* being simply a graphic differentiation of the first curve.

Let the primitive curve A B, Fig. 814, rise from y_1 to y_2 while



the abscissæ change from x_1 to x_2. Then y increases $y_2 - y_1$ units of y for $x_2 - x_1$ units of x, or

$$\left.\begin{array}{r}\text{Rate of growth}\\ \text{of } y\text{, for one unit of } x\end{array}\right\} = \frac{y_2 - y_1}{x_2 - x_1} = \frac{d}{b} = \tan\theta$$

This is the mean rate of growth between a and e, and may be assumed to occur at f, the midway point. The instantaneous rate may be found by supposing b to *gradually* become indefinitely small, when the line $a\ e$ will finally assume a position tangential to curve A B, at the point considered. Hence the rate of growth of a curve is always shewn by the trigonometrical tangent, or the slope, of a line drawn tangential to the curve.

To draw the Sum Curve, that is, to find area A B C D under

any primitive curve, Fig. 815: erect several vertical lines, unequally spaced, there being more where the curve varies greatly, also further mid verticals 11′, 22′, &c. Take any pole O, at a known distance p from a vertical X Y: project points 1′, 2′, 3′, &c., horizontally to X Y, and join each projection to O. Next draw $A a \parallel O1''$, $a b \parallel O2''$, $b c \parallel O3''$, &c., till the whole sum curve A to k be formed, whose ordinate at any point will shew area under primitive curve up to that point, measuring from A; while D j will represent the whole area A B C D.

To Draw the Slope Curve, the reverse process must be performed, and thus the primitive curve in Fig. 815 will shew by its ordinates the slope of the sum curve. For example, suppose we want the slope between $c\,d$, we draw $O4'' \parallel c\,d$, and produce 4″ to 4′, giving a point in what will be the slope curve.

For proof draw $c\,c_1$ horizontally. Then

$$\text{slope of } c\,d = \frac{d\,c_1}{c\,c_1} = \frac{44_1}{p} = \frac{\text{vertical of slope curve}}{p}$$

If we make $p = 1$, the steepness of any primitive curve will be measured by the slope curve ordinate. Again, by cross multiplication,

$$c\,c_1 \times 44' = p \times d\,c_1$$

But $c\,c_1 \times 44'$ is the area between two verticals of primitive curve. Hence if $p = 1$, $d\,c_1$ or rise of sum curve will shew increase of primitive curve area, and ordinate of sum curve at any point will shew area of primitive curve to that point. Also the value p may be so arranged that the new curves may be read off directly to any chosen scale.

Bending Moment and Vertical Shear by Graphic Summation.—The shear on any beam section is due to total load on one side of section less the support reaction (if any) on that side. The beam A C, Fig. 816, is stressed by a load whose intensity per foot run is shewn by curve A B C, drawn to a scale of 100 tons per inch. Base scale being 1 foot per inch or $\frac{1}{12}$, let $\frac{1}{8}$ in. shew 100 tons on shear scale. Supposing an increase of 100 tons on the shear curve, over a base of 1 foot; $d = 100$ tons, $b =$ - in., and y (load) $= 100$ tons $= 1$ in.

But $\dfrac{p}{y} = \dfrac{b}{d} \qquad \therefore\ p = \dfrac{by}{d} = \dfrac{1 \times 1}{\frac{1}{8}} = 8$ in.

and the polar distance is fixed. Taking base D M, summate A B C from pole O, obtaining D E F M, shewing total load on the left of any vertical. Next make D H = M F, and with H as pole summate D E F to base D J, obtaining D G. A vertical through G will pass

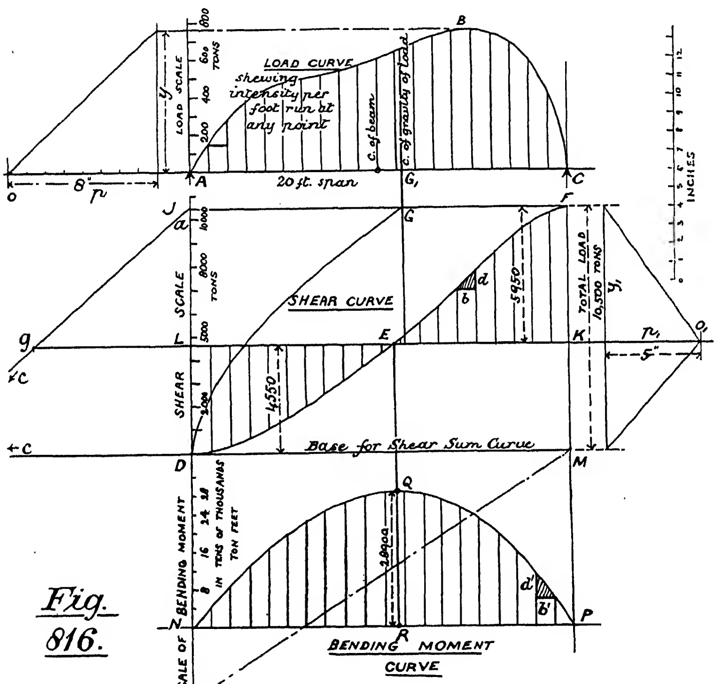


Fig. 816.

Bending Moment & Shear.

through the centre-of-gravity of the load. Set off $a\,c$ = A C, $a\,g$ = A G, and draw g K horizontally. Then D L and F K are right and left reactions respectively. The deduction of area D L K M from D E F M gives the two triangular areas, whose verticals indicate shear, or total load on left of section less left reaction.

From a study of pp. 438 to 442 it will be seen that the B_m curve is simply the sum of the shear curve, the point of origin of the former being where $B_m = 0$, viz., at the supports of a girder, and the outer end of a cantilever. B_m will be a maximum where shear changes sign, and will decrease with minus shear. In Fig. 816 let B_m increase by 1000 ton-feet over a base of 1 foot, caused by a shear of 1000 tons intensity over 1 foot base; and let a B_m scale of $\frac{1}{4}$ in. = 1000 ton-feet be proposed. Then $d' = 1000$ t.f. = $\frac{1}{4}$ in., $b' = 1$ in., $y' = 1000$ tons = $\frac{1}{8}$ in. × 10, and

$$p' = \frac{b'y'}{d'} = \frac{1 \times \frac{1}{8} \times 10}{\frac{1}{4}} = 5 \text{ in.}$$

The B_m curve can now be drawn, on base N P, by summating D L E K F from pole O_1. Commencing at P, the curve rises to Q

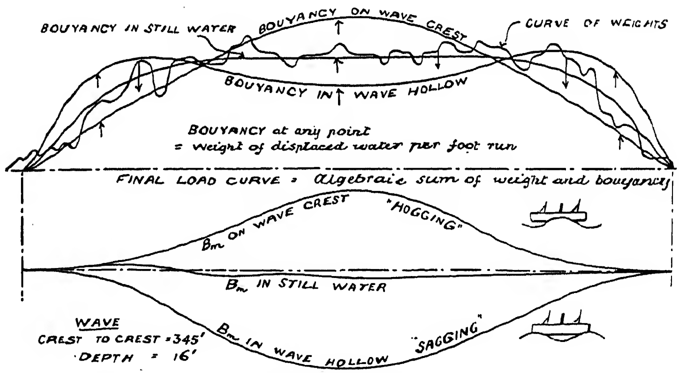


Fig. 817. Ship's Bending Moments.

the maximum, and then decreases from Q to N, the shear being minus. These constructions are especially useful for ships. A curve of weights being drawn, is opposed by a curve of buoyancy or the weight of water displaced at every section, and the net result is the load curve, from which shear and B_m may be deduced as before. Two extreme cases are taken, one with wave crest amidships, causing 'hogging' strains; and the other with crests near the ship ends, causing 'sagging' strains. Lastly the sections are treated as built up beams. Fig. 817 is an actual example.

P. 446. **Culmann's Diagram.**—To prove this construction refer to Fig. 818. Taking any section as B, we shall shew that $B_m = LU \times H$. Reaction x = G J, triangles K L M and O J G are

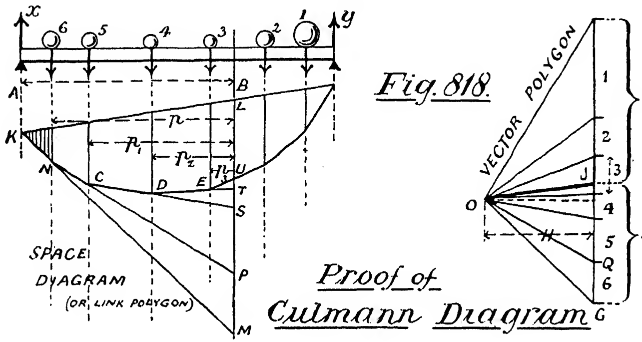


Fig. 818.

Proof of Culmann Diagram

similar, as also are N P M and O Q G; and so on with C S P, D T S, and E U T.

$$\frac{QG}{H} = \frac{PM}{p} \quad \text{and } QG \times p = PM \times H \ldots\ldots\ldots = \text{moment of 6.}$$

$$\text{Similarly } 5 \times p_1 = SP \times H \ldots\ldots\ldots = \text{moment of 5.}$$

$$\text{and } 4 \times p_2 = TS \times H \ldots\ldots\ldots = \text{moment of 4.}$$

$$\text{and } 3 \times p_3 = UT \times H \ldots\ldots\ldots = \text{moment of 3.}$$

$$\text{Again, } \frac{GJ}{H} = \frac{ML}{AB} \quad \text{and } GJ \times AB = ML \times H = \text{moment of } x.$$

$$\therefore \text{Resultant moment} = Mt_x - (Mt_6 + Mt_5 + Mt_4 + Mt_3)$$

$$= ML \times H - (PM + SP + TS + UT)H = \underline{LU \times H.}$$

It is wise to call L U distance and H force, thus keeping space and force diagrams quite separate. Shear is easily proved from previous statements. (*See p.* 1085.)

P. 450. **Deflection of Beams by Graphic Summation.** —Let a beam A V, Fig. 819, be bent to the curve A B C Y, by means of a load of any kind, the beam section being of any shape, but here considered uniform at all sections. Imagine a small portion A B, in which the external fibres D D will be extended or compressed by the amounts d and d_1 respectively; and if δ = strain,

$$d = \delta D = y\frac{d}{y} = y\,\theta$$

Let lines *a* and *b* be tangents to the curve at A and B respectively. Then $\theta = \alpha$, and $\tan \alpha = \alpha$, as α is small. Also $\delta = f/E$.

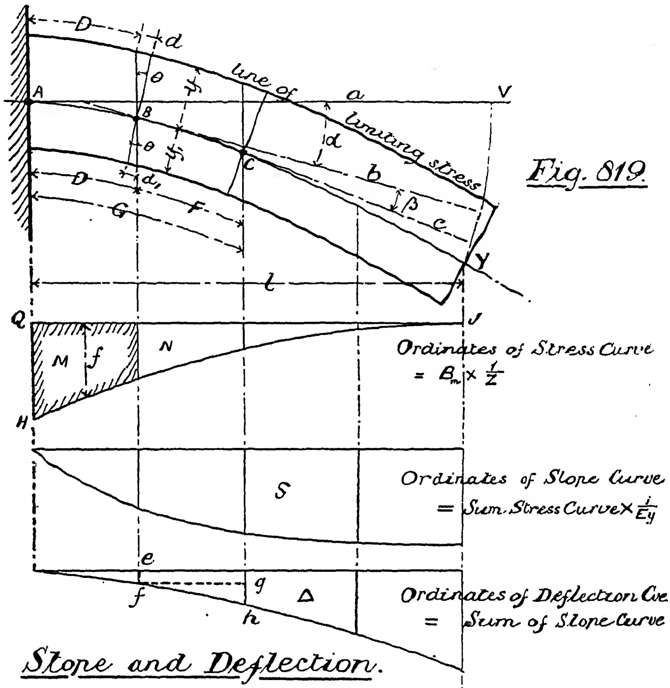


Fig. 819.

Slope and Deflection.

$$\therefore\ \delta D = \frac{f}{E} D = y\theta = y\alpha = y \tan \alpha$$

and $f D = Ey \tan \alpha = Ey$ (slope accumulated from A to B)

Let Q H J be a B_m curve for the load, which we can transform into a limiting-stress curve, for $f = B_m \div Z$. Then

$f D$ = area M = (slope accumulated from A to B) Ey
also, area N = (slope accumulated from B to C) Ey
$\therefore$ areas M + N = (slope accumulated from A to C) Ey.

Hence, $\left.\begin{array}{c}\text{sum of stress}\\ \text{curve to any point}\end{array}\right\} = \left\{\begin{array}{c}\text{total slope at}\\ \text{that point}\end{array}\right\} Ey$

$$\text{And, slope ordinate} = (\text{stress sum-curve ordinate}) \frac{1}{Ey}$$

Thus the slope curve S may be drawn by summating the stress curve, and then dividing by Ey.

Again, if AB be made very small, the angle α = VAB, and tan α = average slope between AB. Then,

$$\text{Deflection at B} = \text{D} \tan\alpha = \text{D (average slope A to B)}.$$

Similarly the angle $(\alpha+\beta)$ = inclination of BC to AV, and tan $(\alpha+\beta)$ = average slope between BC. Then,

$$\begin{aligned}\text{Total deflection at C} &= \text{D} \tan\alpha + \text{F} \tan(\alpha+\beta) \\ &= \text{D (slope AB)} + \text{F (slope BC)} = ef+gh\end{aligned}$$

$$\left.\begin{array}{c}\text{And, sum of slope curve}\\ \text{to any point}\end{array}\right\} = \left\{\begin{array}{c}\text{Total deflection}\\ \text{at that point.}\end{array}\right.$$

The problem is therefore completed by drawing the deflection curve Δ as the summation of slope curve area, and the general formula is deduced for a beam of uniform section:

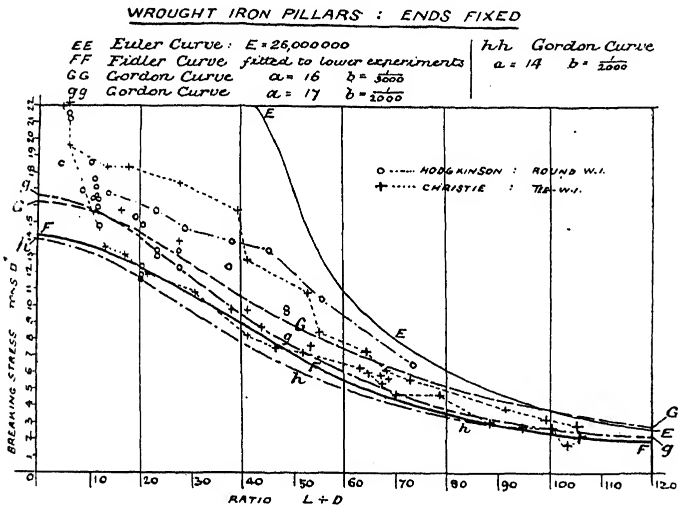
$$\text{Deflection} = \frac{\Sigma(\Sigma B_m)}{Z\,Ey} = \frac{\Sigma(\Sigma B_m)}{E\,I}$$

To prove that the maximum deflection of a cantilever with concentrated load $= W\,l^3/3\,E\,I$ (p. 451): the B_m curve is triangular, whose sum $= W\,l^2/2$, and this is the maximum slope. But the slope curve is a parabola, whose area is therefore $\frac{2}{3}\,W\,l^2/2 \times l$ viz., $W\,l^3/3$, which is the second summation of the B_m; and by above general formula, deflection $= W\,l^3/3\,E\,I$.

Some care is required in fixing scales, but previous explanations may be consulted. When beam section varies, Z will alter also, and the *stress* curve must be found: then continue as before. Note that *load*, *shear*, B_m, *slope*, and *deflection* curves are a continuous series, where each is the sum or integral of the preceding one.

P. 458. **Pillars and Struts.**—In the paper cited on p. 458, Prof. Fidler assigns various reasons why pillar strength cannot be shewn practically by Euler's formula, such as an inconstant E, even in the same strut, and initial curvature in line of thrust, the latter altering W considerably. In Fig. 820 the crosses shew Christie's experiments on **T** bars, and the small circles

Hodgkinson's results for hollow round W. I. columns, co-ordinates being f and r, as on p. 459. As the plottings scarcely approach the Euler curve, the problem is to find what curve will suit. Gordon's curve G with $a = 16$ and $b = \frac{1}{3000}$ evidently strikes an average, which he intended it should. Fidler objects to this treatment, holding that the curve should be made to fit the lower or worst results, and he has therefore devised a formula, too complicated for regular use, and shewn by curve F F. Apparently,



Comparison of Pillar formulæ.
Fig. 820.

however, a judicious alteration of Gordon's constants should approach Fidler's curve, for when $a = 17$ and $b = \frac{1}{2000}$, line gg is drawn; but if $a = 14$ and $b = \frac{1}{2000}$, the lower and safer line hh is obtained. Another way is to take f some 80 or 85% of Gordon, which would meet the experience of engineers.

P. 461. **Combined Torsion and Bending.**—We shall here shew how the two stresses f_t and f_s are combined into one equivalent stress f_e. If a pair of shear stresses F_1 F_1, Fig. 821, act on an imaginary solid A B C D, within a structure, they cannot

exist alone, for they cause a turning effect. They must, therefore, be balanced by a second pair of stresses $F_2 F_2$, where $F_1 \times AB = F_2 \times AD$, or $(f_1 AD)\, AB = (f_2 AB)\, AD$, and $f_1 = f_2$.

Let us next imagine a second block ABCD, Fig. 822, acted on by a shear stress f_s and a direct stress f_t. These may be balanced by the dotted stresses, but we shall consider them resisted by f_e and f_o on plane CB. Now if the value of θ be properly chosen, f_o may be entirely eliminated, and the forces

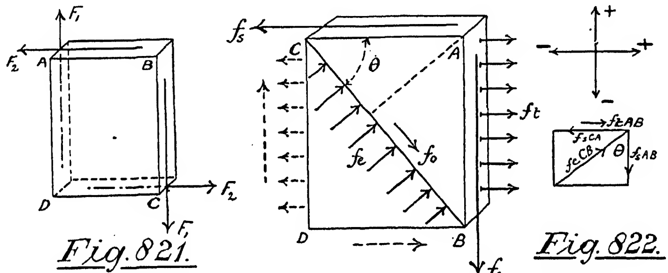


Fig. 821. *Fig. 822.*

Combined Torsion & Bending.

on BC be solely direct, as f_e. Assuming this condition, and resolving f_e CB on CA:

$$(f_e\, CB) \sin\theta = (f_t\, AB) - (f_s\, CA)$$

Dividing by CB: $\quad f_e \sin\theta = f_t \sin\theta - f_s \cos\theta$

$$\text{and } (f_e - f_t) \sin\theta = -f_s \cos\theta \quad \ldots\ldots\ldots\ldots \quad (1)$$

Resolving $(f_e\, CB)$ on AB:

$$(f_e\, CB) \cos\theta = -f_s\, AB$$

Dividing by CB: $\quad f_e \cos\theta = -f_s \sin\theta \quad \ldots\ldots\ldots \quad (2)$

Multiplying (1) and (2) together: $\quad f_e^2 - f_e f_t = f_s^2$

and solving the quadratic: $\quad f_e = \frac{1}{2}\left(f_t + \sqrt{f_t^2 + 4 f_s^2}\right)$

Inserting the values of f_t and f_s on p. 461:

$$f_e = \frac{1}{2}\left(\frac{B_m}{\frac{\pi d^3}{32}} + \sqrt{\frac{B_m^2}{\left(\frac{\pi d^3}{32}\right)^2} + 4\frac{T_m^2}{\left(\frac{\pi d^3}{16}\right)^2}}\right)$$

$$\text{whence} \quad f_e \frac{\pi d^3}{16} = B_m + \sqrt{B_m^2 + T_m^2}$$

which is the so-called equivalent twisting moment, although f_e is a direct stress and not a shear. The result is only true for round shafts, but can be used for other sections by adopting proper values of Z.

CHAPTER IX.

P. 474–8. **Velocity and Energy Curves.** Let any velocity curve, A, Fig. 823, be plotted to a base of equal times; then the acceleration f will be shewn by the slope curve

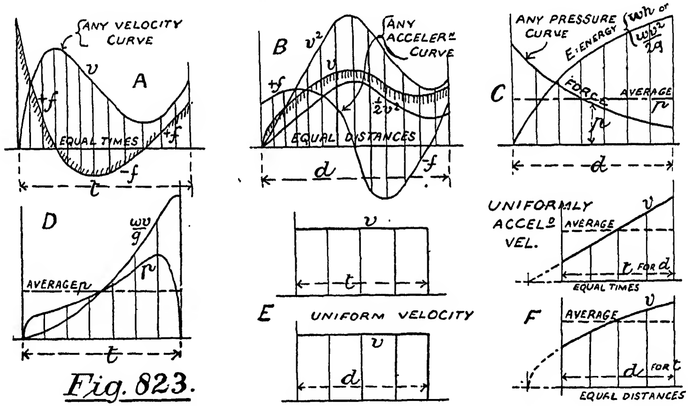


Fig. 823.

Velocity & Energy Curves.

(*see* p. 852) of v, and v will be the sum curve of f, for velocity growth $v = ft$. Hence *acceleration is rate of velocity change regarding time.* Again, at B, if an acceleration curve f be plotted to equal distances, $fd = \frac{1}{2}v^2$ (for $v^2 = 2fd$). Therefore curve $\frac{1}{2}v^2$ is the sum of f regarding d, and the double of these ordinates is v^2, from which v may be found. Conversely, v being known, f may be obtained directly, as on p. 492.

A curve of force at C will give an energy curve by summation, for energy $= p\,d$. Many applications occur: thus, if p be an indicator card, E will shew foot pounds (Wh) given to piston, and

if E shew energy as delivered from a rifle bullet ($Wv^2/2g$) while penetrating a target to depth d, the pressure exerted at any point will be shewn by the p ordinates. Therefore *force is rate of energy change regarding distance.* If, however, the curve p be drawn to a time base as at D, summation will give momentum, or (Wv/g), for momentum = impulse or pt. Then, for a second definition, *force is rate of change of momentum regarding time.* These principles may be carried much further: thus, kinetic energy is the sum of $\frac{1}{2}v$ regarding momentum and so on.

Speaking next of curve averages, it will be easily seen that average velocity, for example, will depend on the base units, and that a time-base average can only equal a distance-base average in the case of uniform velocity, for then distances are proportional to times, and the curves are exactly alike (*see* E). Uniformly accelerated velocity is shewn at F for both time and distance bases, and the average v is evidently less in the former than the latter. In like manner there are time- and distance-base averages of force, as at D and C respectively, a steam-engine-indicator mean pressure being a distance average. (*See p.* 1099.)

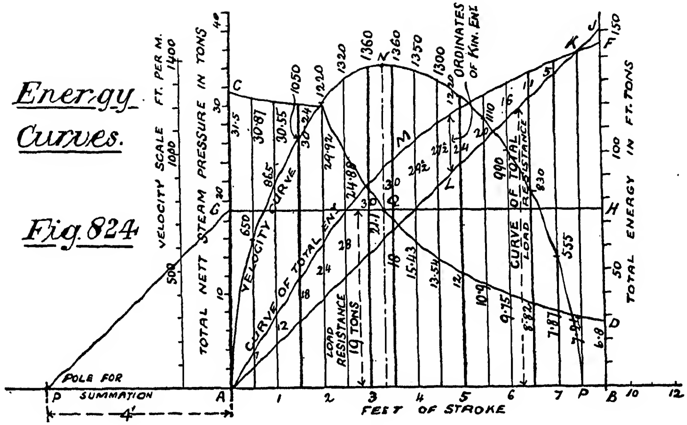
P. 478. Energy Curves.

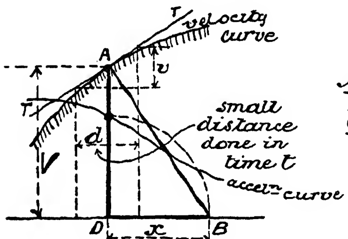
Example 63.—A steam piston of a horizontal engine is 30 ins. diameter, and the net propelling forces due to the steam in lbs. per sq. in., at 6 in. intervals of stroke, are as follows: 100, 98, 97, 96, 95, 79, 67, 57, 49, 43, 38, 34½, 31, 28, 25, 23, 21, &c. The resistances of load and friction are 19 tons, assumed constant, and the weight of the moving parts is taken at 6 tons. Find the correct length of stroke and draw the velocity curve. (Hons. Applied Mechs. Exam., 1897, slightly altered.)

Draw any horizontal base A B, Fig. 824, and set out the 6 in. spaces as shewn. Erect verticals, plot total net steam pressure in tons = p (piston area ÷ 2240) = ·315 p, and draw hyperbolic curve C Q D. As the piston advances from left to right, the area under this curve shews total energy up to a given position. Summate then curve C Q D, and obtain curve A M F of total energy (*see* p. 860). The scale for B F is obtained from previous explanations. Draw G H so that A G = P the total load resistance, = 19 tons. Now, energy absorbed by load is P × D, where D is the stroke swept out up to any point, and the summation of G H, with same pole P, gives A L J, the curve of load energy measurable to same scale as B F. But generally—

Energy Curves.

Fig. 824





Acceleration curve by Proell's method.

Fig. 825.

Ratio of Angular Velocities in Oscillating Engine

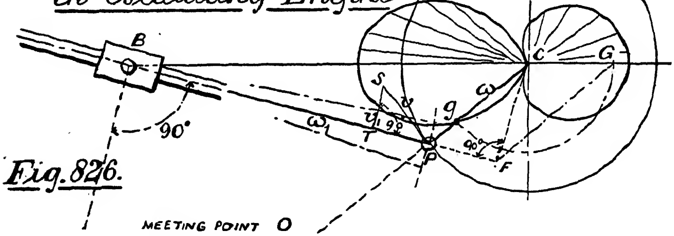


Fig. 826.

Total energy = Potential energy + Kinetic energy, and

$$\text{Total steam energy} = (P \times D) + \frac{wv^2}{2g}$$

where w = weight of moving parts. Hence, as ordinates AMFB shew total energy, and ALJB the load energy absorbed, the remaining ordinates, by deduction, viz., those of AMKL, will indicate kinetic energy of moving parts. Now,

$$\frac{wv^2}{2g} = \text{K.E.} \quad \therefore v = \sqrt{\frac{\text{K.E.} \times 2g}{w}} = \sqrt{\text{K.E.}} \times 4{\cdot}14$$

$$\text{and } V = \sqrt{\text{K.E.}} \times 24{\cdot}8$$

where energy is in foot tons, and $w = 6$ tons. Next, ordinates K.E. are measured on ALK, and velocity curve found by calculation The stroke will finish at P, directly under K, where v equals 0; and the maximum velocity is at N, vertically over Q, where load and steam curves cross. Cut off occurs at $\frac{4}{15}$ of stroke AP.

P. 492. **Acceleration Curves.**—It was shewn at p. 492 how to construct an acceleration curve to a distance base. The proof will here be given by reference to Fig. 825. Let V be a velocity ordinate, whose growth v in a small portion of time t takes place at A; also let a small distance d be traversed during time t. TT is a tangent to the curve, and AB a normal; then DB or x will shew the acceleration f, for

$$f = \frac{v}{t} = \frac{v}{d} \times \frac{d}{t}$$

$$\text{But } \frac{d}{t} = \text{velocity V,} \qquad \text{for space} = tv$$

$$\text{and } \frac{v}{d} = \frac{x}{V} \text{ by similar triangles. Substituting}$$

$$f = \frac{v}{d} \times \frac{d}{t} = \frac{x}{V} \times V = \underline{x}$$

The construction cannot be reversed to find v, but $\frac{1}{2}v^2$ may be found by summation, and v be therefrom deduced.

P. 496. **Comparison of Angular Velocities in Linkwork.**—In pp. 490–496 are found the *linear* velocities of points in linkwork. But it is often convenient to know the ratios of *angular* velocities in a pair of links, and two cases will here

be investigated, as examples of slider-crank and quadric-crank chains respectively.

Fig. 826 is the linkwork of an oscillating engine, where CP is the crank and BP the piston rod, and the ratio ω of CP to ω_1 of BP is required. Linear velocity of P, normally to CP $= v$, and normally to BP $= v_1$. Now if $v = \text{SP}$, $v_1 = \text{ST}$, and

$$\omega_1 = \frac{\text{ST}}{\text{BP}} = \frac{\text{ST}}{\text{BP}} \cdot \frac{v}{v} = \frac{\text{ST}}{\text{BP}} \cdot \frac{v}{\text{SP}}$$

But by similar triangles $\qquad \dfrac{\text{ST}}{\text{SP}} = \dfrac{\text{BP}}{\text{OP}}$

$$\therefore \quad \omega_1 = \frac{\text{BP}}{\text{BP}} \cdot \frac{v}{\text{OP}} \quad \text{and } v = \omega_1 \cdot \text{OP}$$

Again, $\qquad v = \omega \times \text{rad.} = \omega \cdot \text{CP}$

$$\therefore \quad \omega \cdot \text{CP} = \omega_1 \cdot \text{OP}$$

Produce BP to F, and draw CF at right angles to BF.

Also draw GF $\parallel$ CP, making triangles BFG, BPC similar.

Finally $\qquad \dfrac{\omega}{\omega_1} = \dfrac{\text{OP}}{\text{CP}} = \dfrac{\text{BP}}{\text{PF}} = \dfrac{\text{BC}}{\text{CG}}$

or, if BC be angular velocity of crank, CG is that of piston rod Then, if CG be turned round to g, a point is found in a polar curve of angular velocity of BP when ω is constant.

In the quadric crank, Fig. 827, let O be the virtual centre. Then $\omega \cdot \text{AP} = v$, and $\omega_1 \cdot \text{BQ} = v_1$. Draw AL $\parallel$ BQ, making triangles OPQ and APL similar. Also produce QP to S, making triangles SAL and SBQ similar.

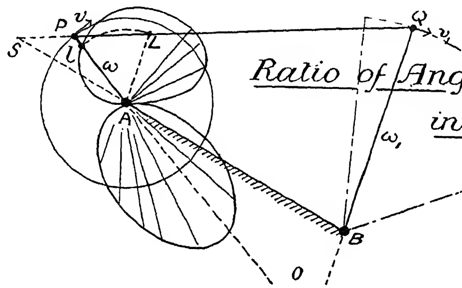
$$\frac{\omega \cdot \text{AP}}{\omega_1 \cdot \text{BQ}} = \frac{v}{v_1} = \frac{\text{OP}}{\text{OQ}} = \frac{\text{AP}}{\text{AL}}$$

$$\text{or} \quad \frac{\omega}{\omega_1} = \frac{\text{AP} \cdot \text{BQ}}{\text{AP} \cdot \text{AL}} = \frac{\text{BQ}}{\text{AL}} = \frac{\text{BS}}{\text{AS}}$$

If the chain be that of a beam engine, ω is constant and represented by BQ, while ω_1 is shewn by AL. Turn AL round on to line AP and a point l is found in a polar curve which shews ω_1 for any position of crank AP.

P. 502. **Efficiency of Transmission by Shafting.—The**

work lost in shafting varies greatly, the limits being 25 to 50% of the power given; but one should distinguish between instantaneous efficiency and that averaged over, say, a whole day.



Ratio of Angular Velocities in Quadric Chain.

Fig. 827.

An extreme case of loss is recorded in Fig. 828, which shews the H.P. curve of an American workshop during one day.

Neglecting engine friction,

		Utilised.	Lost.
Best results for shafting	=	43%	57%
Average results	=	36%	64%

The highest H.P. given to shafting and machines was 44, and the average 39. The machines were next driven by electric motors, with the result shewn, the highest H.P. given to dynamos,

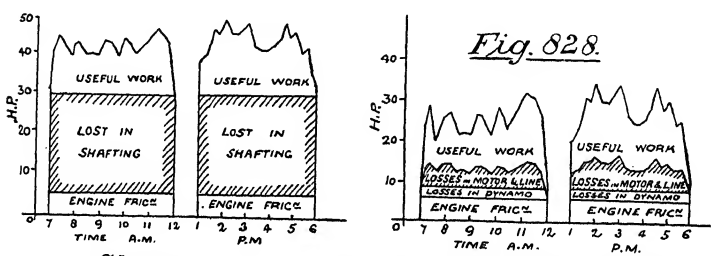


Fig. 828.

Shafting v. Electric Transmission

motors, and machines being reduced to 29 and the average to 21.

Neglecting the engine loss, as before,

		Utilised.	Lost.
Best results for electric driving	=	64%	36%
Average results	=	62%	38%

The advantage of the alteration is self-evident, being largely due to the fact that with electric driving the principal losses vary with the load, and are not constant as with shafting.

P. 504. **Velocity-ratio in Hooke's Joint.**—Given a universal or Hooke's joint, its construction is essentially that of two forks pivoted to a cross, as in Fig. 829. Now let circle P T Q shew plane of motion of P and Q, while P R Q (an ellipse in elevation, but really a circle) indicates that of S and R. Let P move to P_1 while R moves to R_1; then angles P O P_1 and R O R_1

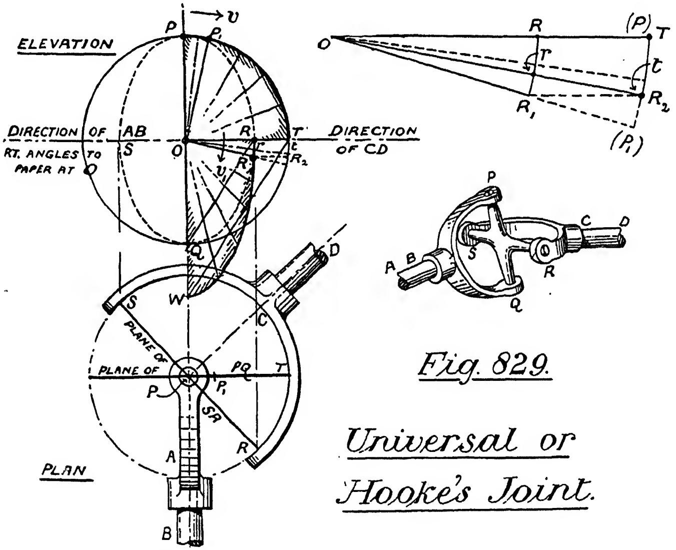


Fig. 829.

Universal or Hooke's Joint.

are equal, the latter being that apparent in elevation and not the real one on plane S R. Draw $R_1 R_2 \parallel$ R T, and join O R_2. T O R_2 is the real angular motion of R, while that of P is P O P_1. Calling P's velocity v, and R's velocity v_1:

$$\frac{v}{v_1} = \frac{P\,P_1}{T\,R_2} = \frac{P\,P_1}{R\,R_1} = \frac{O\,t}{O\,r}$$

When P arrives at T, and R at Q, the velocity relations are exactly reversed, for the fork positions have simply interchanged. Join R Q and draw T W $\parallel$ R Q, thus making O R : O T : : O Q : O W; and complete the quarter ellipse R W. Then the radii vector O P

to O T shew angular velocity of driver A B, taken constant, while O R to O W shew varying angular velocity of follower C D. For, work done being equal, area O P T must equal area O R W, which is only true for circle and ellipse respectively. Thus, area of circle $= \pi$ (O T)^2, and area of ellipse $= \pi$ (O R) (O W), which is π (O T)^2 by construction.

P. 517. **Rolling Curves.**—Other curves may be produced in the manner described at p. 517. Thus, when a parabola is rolled on a straight line, its focus describes a *catenary*, the curve in which a chain hangs, and whose equation is

$$y = \frac{m}{2}\left(e^{\frac{x}{m}} + e^{\frac{-x}{m}}\right)$$

where $e = 2{\cdot}718$, and m is a constant depending on the depth of hang. The ordinate is y and the abscissa x. The *tractrix* or anti-friction curve, is the involute of the catenary.

P. 528. **Tension of Belts.**—Let the arc θ, Fig. 830, of

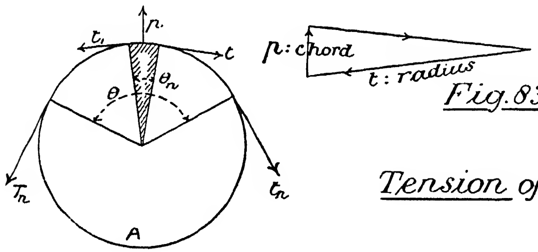


Fig. 830.

Tension of Belts.

the pulley A, be wrapped by a belt, the greatest pull T_n being balanced by $t_n +$ friction. Considering a small elemental strip θ_n, the tensions t_1 and t are balanced by pressure p, and $t_1 = t$ approximately. By force diagram

$$\frac{p}{t} = \frac{\text{chord}}{\text{radius}} = \theta_n \qquad \text{and } p = t\,\theta_n$$

$$\text{Frictional resistance on small arc} = p\,\mu$$

$$\therefore\ p\,\mu = \mu\,t\,\theta_n$$

which is the small increase of tension for arc θ_n. Hence we may write

$$dt = \mu t \theta_n \qquad \text{and} \quad \frac{dt}{t} = \mu \theta_n$$

Summating each side of the equation separately, over the whole angle,

$$\text{Log}_\varepsilon \left(\frac{T_n}{t_n}\right) = \mu \theta$$

But if $\left.\begin{array}{l}\log_b x = a \\ x = b^a\end{array}\right\}$ $\therefore \frac{T_n}{t_n} = \varepsilon^{\mu\theta}$ and $\text{Log}_{10}\left(\frac{T_n}{t_n}\right) = {\cdot}4343\,\mu\,\theta$

P. 555. **Friction.**—Experiments on solid friction may be classified under six heads:—

1. Static friction.
2. Friction at low speeds.
3. Friction at high speeds.
4. Journal friction.
5. Collar friction.
6. Pivot friction.

And in every case the surfaces may be wholly or partially lubricated, may be dry, or may be coated with a resistant. The first three are usually taken with flat surfaces, and the results known as 'flat friction,' unit pressure being constant over the surface. In the remaining cases the surface pressure may be very unequal.

The friction at starting, called friction of repose or static friction, has been but little examined, but some good results were obtained in 1895 by Mr. Broomall, of America, being the averages of twenty or thirty trials with each load.

VALUES OF μ FOR STATIC FRICTION.

Material.	Dry.	Wet.
Cast iron on cast iron	·3114	·3401
Steel on steel	·4408	
Steel on cast iron	·2303	
Cast iron on tin	·4541	
Steel on tin	·3648	
Cast iron on pine	·4702	
Pine on pine	·4738	·6350

The extreme cases were from ·96 to 1·04 times these numbers, but at certain loads the values became constant. As p increased μ decreased, except with dry pine on pine, when the reverse occurred. The effect of long contact was to slightly increase μ with the dry experiments, and markedly so with wet pine on pine.

For low-speed friction, Morin's results may be accepted. The method of experiment is to find a load P which will just keep a weight W moving at a slow uniform speed on a level

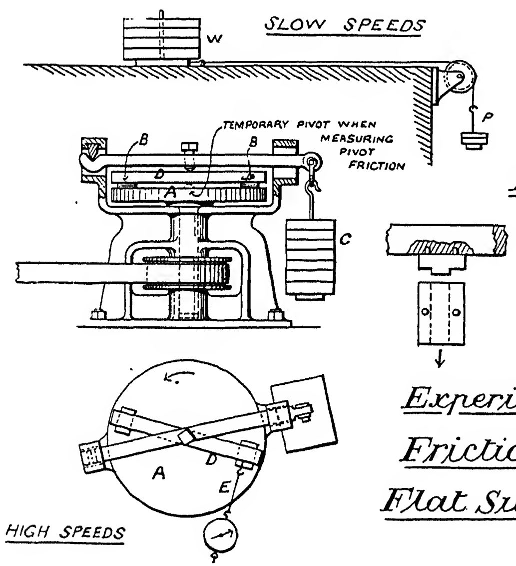


Fig. 831.

Experiments on Friction of Flat Surfaces.

surface, Fig. 831: then the ratio $P:W=\mu$, and very regular results are obtainable.

At high speed somewhat conflicting figures are found. Apparently there is a decrease in friction with increase of speed if the surfaces are dry (p. 556), or rather, at high speeds the friction seems to approach the lubricated cases, most probably due to a cushion of air drawn in between the surfaces. High-speed lubricated experiments are troublesome, because the

lubricant alters with the temperature, every oil having a best condition, but the results of flat friction at linear velocities from 400 to 1600 ft. per m. shew a constant μ of ·23 at 50 lbs. per sq. in.: for cast-iron surfaces. The apparatus is shewn in Fig. 831, where A is a rotating disc, and B B two rubbing surfaces (shewn in detail on the right) pressed upon it by the load C. The turning effort of D is resisted by string E, and F_n measured by spring-balance or scale-pan weights.

Beauchamp Tower's journal experiments for the Inst. of Mech. E. were made at high speeds with heavy loads, and the projected area ld adopted for measurement of unit pressures, d being width of journal embraced, and l its length. The nominal p was thus conventional, the real p often rising to twice its amount. The load was carried on a knife-edge K, Fig. 832 and C K would assume some position C L. Then

$$\text{Moment of friction} = \text{Moment of weight}$$
$$F_n r = Pa$$

The movement was multiplied by levers A and B, but still being small, was measured by weights in a scale-pan D sufficient to bring B back to zero. The starting stickiness was overcome by a small weight E, and the lever thus kept horizontal for zero load, the amount being decided by revolving in each direction and taking the average. Briefly, total friction was found constant at all loads, and μ varied as $1 \div P$. Oil-bath lubrication proved best, sponge lubrication some four times as resistant, and siphons gave still worse results. Temperature materially altered the friction; thus by lowering from 300° to 120° Fahr. the friction with lard oil was as 1 : 3. By means of pressure gauges it was shewn to be useless to introduce the lubricant where pressure was greatest, for the latter often rose to 200 lbs. per sq. inch and forced out the oil. Oil should therefore enter at the top of shafts and the bottom of axles, and grooves be cut to assist it. (*See Third Preface.*)

With collar bearings (Inst. Mech. E. experiments) the friction varied more nearly with the pressure, the apparatus being shewn in Fig. 832*a*. A ring C supported on rollers J J, was pressed

between discs A and B by the force of a screwed-up spring H, the amount of pressure being measured by pointer H, and the bolt F having a spherical seat. Both discs being connected by keys K K,

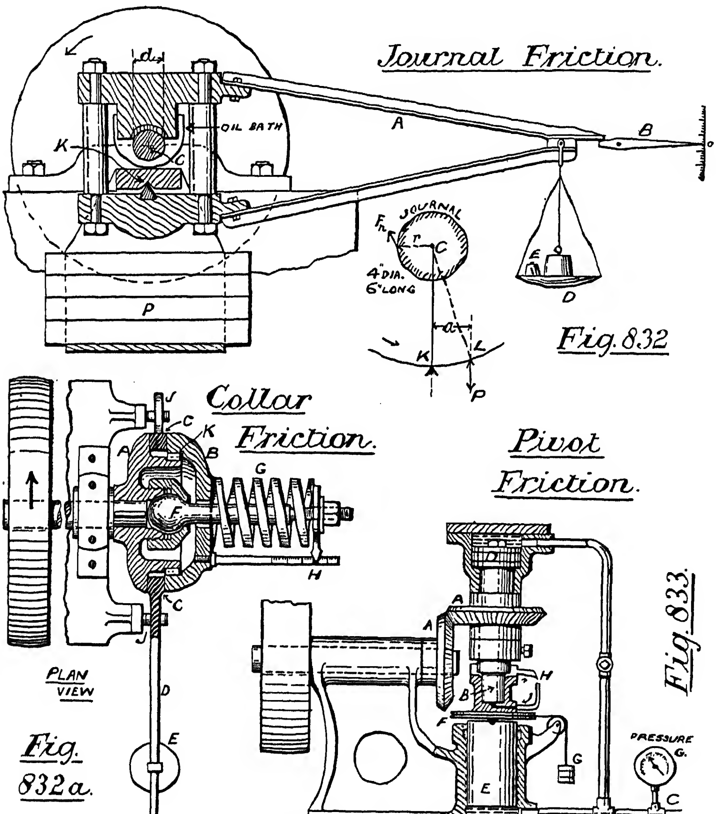


Fig. 832

Fig. 832a.

Fig. 833.

are revolved by belting, and the friction between the two surfaces is measured by weight E, in the manner of a dynamometer (see also p. 558). In marine practice pressures of 50 lbs. per sq. in. are allowed on thrust-block collars, and the H.P. transmitted to

the screw taken at $\frac{2}{3}$ the I.H.P. The direct thrust may be found from speed, for

$$V \text{ of ship} = \text{knots} \times 101{\cdot}3$$

$$\text{Thrust} \times \text{knots} \times 101{\cdot}3 = \text{effective H.P.} \times 33000$$

$$\text{or } P = \frac{(\text{eff. H.P.}) \times 33000}{K \times 101{\cdot}3} = 217 \frac{\text{I.H.P.}}{K}$$

$$\therefore \text{ Collar surface required} = \frac{217 \text{ I.H.P}}{50 \text{ K}} = 4{\cdot}34 \frac{\text{I.H.P.}}{K}$$

It must be noted that these surfaces are now horseshoe in form (see p. 691) and more collars are required than if circular.

For pivot bearings the experimental apparatus in Fig. 833 was adopted, where B is the pivot or footstep fed by oil entering at pipe H. The shaft D being rotated through bevel gearing A A, the frictional moment was measured by weight G, which, acting on pulley F, prevented the bearing at B from rotating. At the same time, the load was obtained by oil pumped against surfaces D and E, its intensity being measured by pressure gauge C.

P. 568. **Balls and Live Rollers.**—It is found that with ball or roller bearings the frictional loss is $\frac{1}{6}$ or $\frac{1}{7}$ of that of a plain journal, and in large bearings the rollers are kept apart by rings, as in Fig. 584.

P. 575. **Efficiencies of Machines.**—The example on pp. 571 to 575 shews the methods usually followed in mechanical laboratories to find frictional loss in machines. Two further cases may be given by way of illustration. Fig. 834 is a chart of experiments on rope-pulley blocks, a simple fixed pulley being called a 1 : 1 system, one movable and one fixed block a 2 : 1 system, and so on. Thus 3 upper and 3 lower pulleys make a 6 : 1 system. Plotting P : W the inclined lines are obtained, and efficiency curves are further calculated for each load. A curve of *maximum* efficiency is also drawn below, on a base of velocity-ratio, which usefully indicates the rapid loss with increased theoretical advantage. The sheaves were $2\frac{1}{2}''$ diameter and the rope $\frac{1}{2}''$.

Investigating, we have in a 1 : 1 system :

$P = W(1 + m)$ where m is a proper fraction.

Then, in a 3 : 1 system, say:

$$P_1 = P_2(1+m) \ldots\ldots P_2 = P_3(1+m) \ldots\ldots P_3 = P_4(1+m) \quad \ldots\ldots(1)$$

Adding each side*:

$$P_1 + P_2 + P_3 = (P_2 + P_3 + P_4)(1+m)$$

$$\text{or} \quad P_1 = m(P_2 + P_3 + P_4) + P_4$$

But since all the rope tensions together = W

$$P_2 + P_3 + P_4 = W$$

$$\text{and} \quad P_1 = P_4 + mW \quad \ldots\ldots\ldots\ldots\ldots\ldots(2)$$

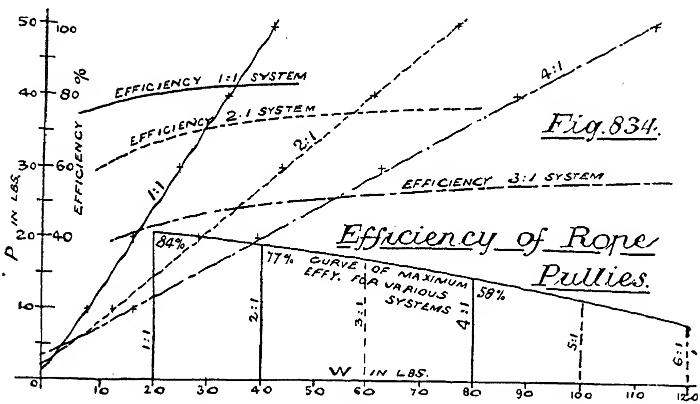


Fig. 834.

Efficiency of Rope Pullies.

Again, multiplying sides together in (1):

$$P_1 P_2 P_3 = P_2 P_3 P_4 (1+m)^3 \qquad \therefore\ P_1 = P_4(1+m)^3$$

$$\text{and} \quad \frac{P_1}{(1+m)^3} = P_4 \quad \ldots\ldots\ldots\ldots\ldots\ldots(3)$$

Substituting this in (2)

$$\left.\begin{aligned} P_1 &= \frac{P_1}{(1+m)^3} + mW \\ \text{or} \quad P_1\left(1 - \frac{1}{(1+m)^3}\right) &= mW \end{aligned}\right\} \text{for a } 3:1 \text{ system} \ldots\ldots(4)$$

$$\text{or generally} \qquad \frac{W}{P_1} = \frac{1 - \frac{1}{(1+m)^n}}{m}$$

* P_1, P_2, P_3 are the rope parts taken in order from P to W. See Fig. 439.

where the exponent n represents the number of cords; and the *real* advantage is thus shewn. The assumption, however, that pulley friction and rope-bending resistance vary similarly is not quite true, the latter increasing more rapidly than the former. Comparing with experiment,

$$1:1 \text{ block}; \quad m = \frac{P-W}{W} = \cdot 195 \quad \text{at max. effy}$$

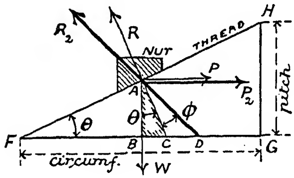
$$2:1 \text{ block}; \quad \frac{W}{P} = \frac{1 - \dfrac{1}{(1+m)^2}}{m} = 1\cdot 54$$

which, by experiment = 1·52 at max. effy.

$$4:1 \text{ block}; \quad \frac{W}{P} = \frac{1 - \dfrac{1}{(1+m)^4}}{m} = 2\cdot 62$$

which, by experiment = 2·25 at max. effy.

In the case of a screw jack, the P : W line passed through the



Efficiency of Screw.

Fig. 835.

origin, P & W vanishing simultaneously, because the screw was reckoned as load. The efficiency was therefore constant at all loads, being ·3535 with a screw 1·92″ mean diameter and pitch ½″, the velocity ratio being 59·33 : 1. The lever was represented by a pulley mounted on the screw axis. Here again the efficiency can be found by calculation. Let F G, Fig. 835, be the mean screw circumference, and H G the pitch, the nut being moved by force P. Neglecting friction, there are three balancing forces, P, R, and W; and the force diagram is A B C.

$$\frac{W}{P} = \frac{AB}{BC} = \frac{\text{circumference}}{\text{pitch}} = \frac{1}{\tan\theta}$$

With friction, the real resistance is R_2, and the force diagram A B D, where ϕ = friction angle.

$$\frac{W}{P_2} = \frac{A\,B}{B\,D} = \frac{1}{\tan(\theta+\phi)}$$

$$\text{and efficiency} = \frac{P}{P_2} = \frac{\tan\theta}{\tan(\theta+\phi)}$$

shewing that large pitch is economical. Returning to the experiment,

$$\tan(\theta+\phi) = \frac{\tan\theta}{\text{effy.}} = \frac{\cdot 083}{\cdot 3535} = \cdot 235$$

$$\therefore \phi = (\theta+\phi) - \theta = 13{\cdot}27^\circ - 4{\cdot}76^\circ = 8{\cdot}51^\circ$$

$$\text{and} \quad \mu = \tan\phi = \underline{\cdot 1494} \qquad (\textit{See p. } 1125.)$$

P. 576. **Absorption Dynamometer.**—The apparatus in Fig. 596 is called an Appold brake. If P = pull on stud D, r = radius of brake wheel, and F_n total friction on brake strap, the sum of moments being necessarily zero,

$$(W - S)\,R \pm (P \times A\,D) = F_n\,r$$

or the total moment exerted by the engine. P may be measured by two spring balances, one on each side of D, a pull on the right balance being plus, and on the left minus. The work absorbed per revolution would be

$$2\pi\left\{(W - S)R \pm (P \times A\,D)\right\}$$

whence B.H.P. is found. If the H.P. be under 15 and the lubrication sparing, there is little pull on D, but the lever is generally a bad arrangement, and a simple strap is now advised, where only W and S are measured.

P. 580. **Distribution of Power.**—We may distinguish between mere transmission, and distribution from a central station to many consumers. Professor Unwin has shewn the advantages of the latter over individual installations, and gives the following requirements:

1. Indefinite subdivision and measurement.
2. Minimum first cost and running loss.
3. Freedom from danger.

4. Consumer's motor to be simple and efficient.
5. Facility for adaptation to numerous uses.

Only two great natural sources of power have been much used up to the present, gravity and heat, the former being the water power of streams, and the latter the latent energy of coal or petroleum. The distributors have been electricity, wire rope, compressed air, hydraulics, or coal gas; of which probably compressed air meets the above requirements most perfectly. Intermittent use means the necessity for storage if a part of the plant is not to lie idle at times, and electric accumulators are very costly for large stations, but hydraulic power may be efficiently stored. A low-pressure water power is used in Switzerland, drawn from reservoirs 400 ft. high kept filled by turbines, and the storage of heat has been accomplished by admitting surplus boiler steam to a large vessel of water under high pressure, the liquid thus heated being converted into steam when needed, at a small reduction of pressure.

CHAPTER X.

P. 583. **Conduction of Heat through Plates.** — The boiler furnace having a temperature of about 1500°, and the steam of 300°, it may be taken that there is a 'skin' drop of 500° on each side, and another drop of 300° in passing through the plate.

P. 587. **Le Chatelier's Pyrometer** is undoubtedly the finest modern apparatus for measuring high temperatures, and, as improved by Roberts-Austen, is shewn in Fig. 836. A is a thermo-electric couple of two wires, twisted together, one of platinum and the other of platinum alloyed with 10% of rhodium. Being protected with refractory material, they are placed in a source of heat, and connected to the mirror galvanometer B, within the camera C. A ray of light from an oxy-hydrogen burner D is deflected and focussed on to the mirror E, from which it returns to a photographic plate F. Now the horizontal movement of the light ray at F will indicate temperature: but a better way is to cause the plate F to move vertically by clockwork while the heat changes occur at

A, thus recording a time-temperature curve. The temperature scale is graduated by reference to well-known phenomena, whose

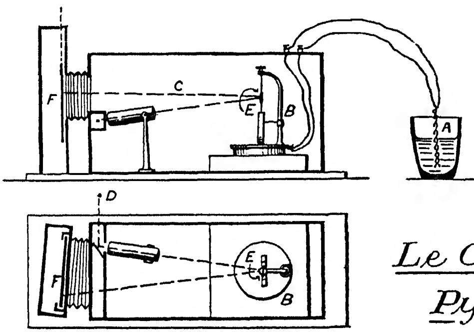
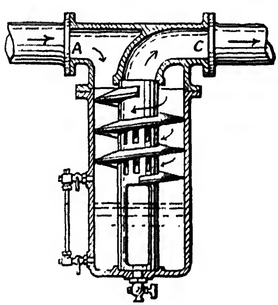


Fig. 836.

Le Chatelier's Pyrometer.

temperatures have been accurately obtained by the methods on p. 587, and the junction at A should directly touch the source of heat.

P. 593. **Steam Drier or Separator.**—This apparatus is now considered essential to an engine steam-pipe, especially if the



Steam Drier or Separator.

Fig. 837.

length of the latter be great, and the principles embodied in various forms may be illustrated by Fig. 837. Steam enters at A, and in passing round the helix to escape at C, deposits all the moisture on the casing wall by centrifugal action, only dry steam passing away.

P. 594. **Dryness of Steam.** — The finding of dryness value from indicator diagram has already been shewn at p. 764. Professor Carpenter plots a further curve beneath the expansion line to represent the fractions thus obtained, and calls this the 'quality' curve of the steam. It appears that condensation continues after cut-off A, Fig. 838, till re-evaporation is reached at B,

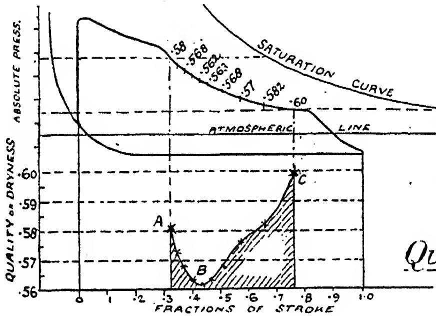


Fig. 838.

Dryness- or Quality-curve.

while the final dryness C is often higher than A. To measure the condensation due to cylinder walls we must know the original condition of the steam. Several methods are used, but Carpenter's apparatus, Fig. 839, is probably simplest, while being very

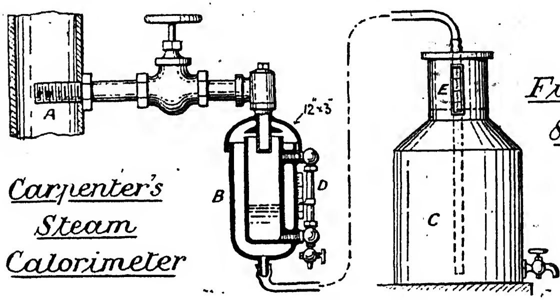


Fig. 839.

Carpenter's Steam Calorimeter

accurate. Steam passes from the pipe A into a jacketed separator B, where all the moisture is deposited. Continuing to the water vessel C, the dry steam is there condensed, and the graduations

at E and D will respectively shew weights of dry steam and entrained water.

P. 605. **Problems on Energy Changes in a Working Gas.**—When a gas is expanded or compressed in any manner behind a working piston,

Heat supplied = change of internal energy + external work

and any of the three terms may be plus, minus, or zero. Thus, when expanding isothermally, the internal change is zero, and the gas must be given a positive heat equal to the total work done (p. 605); and in isothermal compression the heat supply is negative, that is, must be abstracted to the same amount as before.

Example 64. Assuming the pv curve to be hyperbolic, let steam enter the cylinder at 120 lbs. absolute pressure per square in., temperature 348° F., and expand to 30 lbs. absolute, temperature 250° F. Find heat supplied per lb. weight. (Hons. Steam Exam. 1894.)

Spec. vol. at 250° = 13·5 cub. ft. and P = (144×30) lbs.

$$\therefore \quad PV = 144 \times 30 \times 13{\cdot}5 = 58320 \text{ ft. lbs.; which is constant.}$$

$$r = \frac{V_2}{V_1} = \frac{P_1}{P_2} = \frac{120}{30} = 4 \qquad \text{and } \log_e r = 1{\cdot}3863$$

$$\text{Total heat at first,} \quad H_1 = S_1 + L_1 = 348 - 32 + 871 = 1187 \text{ B.T.U.}$$

$$\text{Total heat at end,} \quad H_2 = S_2 + L_2 = 250 - 32 + 940 = 1158 \text{ B.T.U.}$$

$$\text{Loss of internal energy} = 772\,(H_1 - PV) - 772\,(H_2 - PV)$$
$$= 22388 \text{ ft. lbs.}$$

$$\therefore \quad \text{Heat supplied} = \text{external work} - \text{internal change}$$
$$= PV \log_e r - 22388 = (58320 \times 1{\cdot}3863) - 22388$$
$$= 58461 \text{ ft. lbs., or } \underline{74{\cdot}7 \text{ B.T.U.}} \text{ per } 13\tfrac{1}{2} \text{ cub. ft., or per 1 lb. weight.}$$

[N.B.—The real result should be 87 B.T.U., the discrepancy being due to assuming the expansion of dry steam as hyperbolic.]

Example 65.—Find expressions for the thermal efficiency, when atmospheric air at 60° Fahr. is compressed adiabatically, then cooled under constant pressure to 60° Fahr., and afterwards exhausted adiabatically while doing work, till it reaches the atmospheric pressure again.

Referring to Fig. 840, O D and O F are scales of absolute pressure and volume respectively. Air is drawn in along A B, compressed to C,

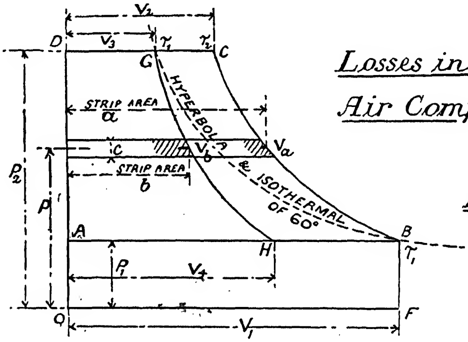
and discharged along C D, the pressure again falling to A. The work done *on* the air is the area A B C D. Similarly, in the motor, we admit along D G, expand from G to H, and discharge along H A, the pressure rising again to D. The work done *by* the air is D G H A.

$$\therefore \quad \text{Efficiency} = \frac{\text{work returned}}{\text{work expended}} = \frac{\text{D G H A}}{\text{A B C D}}$$

and these areas are to be measured. For any pressure P, $PV^{\gamma} = C$

$$\therefore \quad \frac{P\,V_a{}^{\gamma}}{P\,V_b{}^{\gamma}} = \frac{C_a}{C_b} \qquad \text{and } \frac{V_a}{V_b} = \text{a constant}$$

which is true anywhere on the adiabatics. Imagine the areas cut into horizontal strips, each of a very small width c, and let a = area of



Losses in Air Compression.

Fig. 840.

one strip to V_a, while b is that to V_b. Then $a = c\,V_a$ and $b = c\,V_b$. Sum the strips for the whole areas, and let $V_a = R\,V_b$.

$$\frac{\Sigma b}{\Sigma a} = \frac{c\,\Sigma V_b}{c\,\Sigma V_a} = \frac{\Sigma V_b}{\Sigma V_b R} = \frac{1}{R} = \frac{V_b}{V_a}$$

$$\therefore \quad \frac{\text{Area D G H A}}{\text{Area A B C D}} = \frac{V_b}{V_a} \qquad \text{and } \eta = \frac{V_3}{V_2} \text{ or } \frac{V_4}{V_1}$$

$$\text{Now suppose ratio of compression } \frac{V_1}{V_2} = r$$

$$\text{Then by p 607 } \frac{\tau_1}{\tau_2} = \left(\frac{1}{r}\right)^{\gamma - 1}$$

But, by changing from τ_2 at C to τ_1 at G by constant pressure, we have, by Charles' law:

$$\frac{V_3}{V_2} = \frac{\tau_1}{\tau_2} = \left(\frac{1}{r}\right)^{\gamma - 1} \qquad \text{and } \eta = \left(\frac{V_2}{V_1}\right)^{\gamma - 1}$$

Again, suppose the air to be compressed from 1 to x atmospheres,

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^{\gamma} \qquad \text{Raising to the } \frac{\gamma-1}{\gamma} \text{ power,}$$

$$\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} \qquad \therefore\ \eta = \underline{\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{x}\right)^{\frac{\gamma-1}{\gamma}}}$$

We may deduce some important results from the last two xamples. If a gas be compressed or expanded in any manner, nd then returned to the original temperature, the heat supplied r removed must equal the work area as indicated, for the internal nergy is then unchanged. Fig. 841 is a general diagram for air-

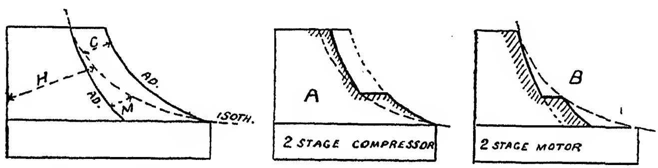


Fig. 841. Air-Compression Systems.

ompressor and motor, and we may take three cases:—

$$\text{I.}\begin{cases}\text{Adiabatic compression: cooling back to isothermal:} & \\ \qquad\qquad\qquad\qquad\text{heat lost...} & = H+C \\ \text{Adiabatic expansion: work gained (from gas)} & = H-M\end{cases}$$

$$\therefore\ \text{Total loss in system} = (H+C)-(H-M) = C+M$$

$$\text{II.}\begin{cases}\text{Isothermal compression: no cooling: heat lost} & = H \\ \text{Adiabatic expansion: work gained ...} & = H-M\end{cases}$$

$$\therefore\ \text{Total loss in system} = H-(H-M) = M$$

$$\text{III.}\begin{cases}\text{Isothermal compression: heat lost...} & = H \\ \text{Isothermal expansion:}\left\{\begin{matrix}\text{heat to be supplied} = H \\ \text{work gained } \ldots = H\end{matrix}\right\}\text{loss} & = 0\end{cases}$$

$$\therefore\ \text{Total loss in system} \ldots = H$$

The isothermal line is most closely approached by two-stage ompressors (p. 547), and by two-stage motors, whose diagrams re given at A and B respectively, Fig. 841; but in the second it must be understood that all the external work must be supplied

by re-heaters, the only reason for whose success lies in the remarkable economy of heat thus supplied, some six times as efficiently as when used in a steam boiler.

P. 605–8. **Plotting the Curve $pV^n = C$.**—The pressure for given volume, or volume for given pressure, can be found by using a table of logarithms, for

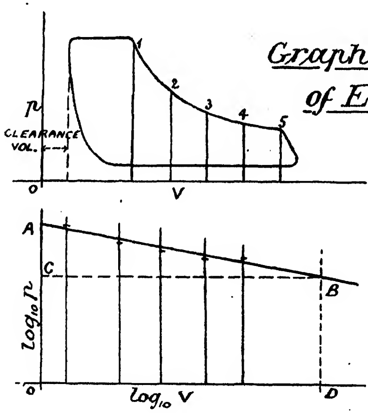
$$\log p + n\log V = \log C$$

$$\therefore \quad \log p = \log C - n\log V$$

$$\text{and } \log V = \frac{\log C - \log p}{n}$$

[Note that common logs. are used.]

The values p or V may then be deduced, being the numbers corresponding to their logarithms. Or, again, the values n and C for a given curve, may be found by construction. Taking the given expansion curve, Fig. 842, measure every pressure and



Graphic determination of Expansion-curve Equation.

Fig. 842.

volume at any points 1, 2, 3, 4, 5, and, on a new diagram, shewn below, plot their logarithms, drawing the best straight line A B through the plotting. Then

$$n = \frac{\text{A C}}{\text{C B}} \qquad \text{and } \log p + \frac{\text{A C}}{\text{C B}} \log V = \log C$$

Putting in the values for $\log p$ and log V, taken anywhere, log C

is obtained, and C, the corresponding number, is taken from the tables. (*See pp.* 1131 *and* 1168.)

P. 611. **Thermal Efficiency of Steam Engines.**—It is now practically agreed that engine efficiency should be measured apart from the boiler, and the two efficiencies multiplied if the whole economy be required. There having been some disagreement as to measurement of the first, a committee formed in 1897 by the Inst. C.E., after consideration, reported that the efficiency of a steam engine should be stated by

I. Number of B.T.U. per I.H.P. per hour:
II. Number of B.T.U. per B.H.P. per hour:
III. Comparison of heat consumption with that of an ideal cycle, and called 'Comparison ratio' (see rel. effy. p. 772):

all three being given if possible. The B.T.U. are to be measured by taking steam temperature before entering engine stop-valve, and again when entering exhaust pipe; then reckoning the heat required to raise the weight of steam used, from water at exhaust to steam at initial temperature, under constant pressure. The ideal cycle for steam is to be taken as the 'Rankine' cycle, which is only different from Carnot, p. 772, in having no compression, and being therefore non-reversible. There is little difference between the two when using saturated steam, but with superheated steam the Rankine efficiency is much less. As the Carnot formula is much simpler, it may be used for approximation in many cases. Then

$$\text{Comparison ratio} = \frac{\text{B.T.U. by Rankine or Carnot}}{\text{B.T.U. actually used.}} \quad (\textit{See p. } 1168.)$$

P. 612. **Reversible Cycles.** Cotterill shews that when the exhaust steam is used to heat the boiler feed, the cycle is much more nearly reversible. Further, that if Weir's system of feed heating in stages be adopted, by steam drawn from the intermediate receivers of multiple-stage compounds, a still better cycle is obtained, reaching ideal conditions if the number of stages be infinite. The method is then exactly similar to Stirling's, p. 771, the receivers acting as regenerators or temporary heat stores.

P. 614. **Calculation of Initial Condensation and Leakage.**—The weight of steam used in addition to that shewn by the indicator is a loss spoken of as the 'missing quantity,' and is ascribed to cylinder condensation and leakage. The researches of Hirn, D. K. Clark Willans, Cotterill, Thurston, Donkin, Adams, English, and Callendar have been directed towards the rationalisation of this loss, in order to predict it in any case where the conditions of pressure, speed, clearance, expansion, &c., are known. Donkin, Adams, and Callendar have experimentally obtained temperature cycles of the steam and the cylinder walls, which fully shew the interchanges that have been generally admitted. They also shew that steam temperatures may be accurately deduced from the indicator diagram by converting pressures into corresponding temperatures. Cotterill devised a semi-empirical formula, based on barrel surface, as follows:—

$$\text{Total steam consumption} = \text{weight of dry steam} \times \left(1 + \frac{C \log_e r}{d\sqrt{N}}\right)$$

where d = cylinder diameter. Callendar's experiments appear to shew the greatest loss to be due to clearance surface, while leakage may be responsible for from 5 to 50% of the total loss. The constant C is about 6 to 8 in simple unjacketed engines, and 4 in jacketed slide-valve engines, 2 to 4 in Corliss engines depending on the presence or absence of jackets, and 12 in very bad engines; but rarely rises above 8. Cotterill calls the bracketed terms 'liquefaction factor,' and another method of statement is

$$\frac{\text{Missing steam}}{\text{Indicated steam}} = \frac{C \log_e r}{d\sqrt{N}}$$

The missing quantity may vary from 25 to 60% of total steam supplied, and is about 30% on the average. Fig. 755 shews the loss graphically. (*See p.* 1134.)

P. 614. **Methods of Reducing Condensation.**—There are four methods of decreasing the loss just mentioned: quick running, steam-jacketing, superheating, and compounding.

Quick running.—The advantage of high rotational speed (not

necessarily high piston speed) was demonstrated by Mr. Willans in 1888, when he got very economical results with a simple engine greatly by that means. The time of each stroke is then too short to allow of the temperature changes which cause condensation loss, and the expansion is approximately adiabatic Lagging the cylinder increases its non-conducting capacity, and is an additional help.

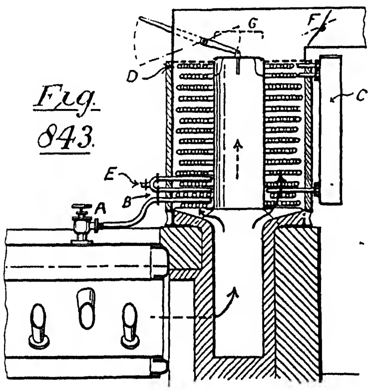
Steam-jacketing is different to lagging in that heat is actually given to the steam, to assist re-evaporation. In most cases there is some advantage in its use, though not always a large one. Its good effect is greatest with simple engines, slow speeds, large rate of expansion, and wherever considerable condensation would otherwise occur: it is of less value in compounds and in non-condensing engines, while at high rotational speeds it is useless.

Superheating the steam, that is raising its temperature above saturation point before entering the cylinder, is the most direct and effective means of reducing and even eliminating condensation loss. This is not achieved by increased boiler heating surface, nor by improved thermal efficiency due to higher temperature, both of which false explanations have been made; but simply by supplying the saturated steam with such heat as will partly or entirely prevent condensation, thus keeping the steam dry, or nearly so, throughout the stroke. The curious anomaly occurs that a 5% extra heat supply will often eliminate a 20% loss. Hirn demonstrated a saving of 22% in 1855, and in 1859 Penn and others saved as much as 30%; but as steam pressures increased, a reasonable 'superheat' could not be obtained without danger at the superheater, and the cylinder lubricants were also burned up, so the process was abandoned in 1870. A revival has taken place since 1890, the original objections having been removed by the substitution of petroleum for animal oils, and safer superheaters having small parts. The gain varies from 10% to 50% with 50° to 100° superheat respectively, and according to the perfection of the engine itself; but averages 25%. The effect on the indicator diagram is to raise the expansion curve, while lowering the saturation curve due to feed, and thus the dryness fraction approaches a maximum.

Fig. 843 shews the Schmidt superheater, consisting of a coi

of pipes through which the steam passes, and placed as near the boiler as possible. Leaving the boiler at A, the steam enters the two lower coils at B, then goes to the 'after-evaporator' C, and the uppermost coil D, from which it makes its way coil by coil to leave at E for the engine. In this way wet steam circulates where the heat is greatest, while the main and final superheat are more

Fig. 843.



Schmidt Superheater.

effective by placing steam and gas currents in contrary directions. The damper G adjusts the superheat, which is a maximum when G is closed, and nothing when open. This important attachment thus meets changes in steam demand or stoking variation.

Compounding was explained at pp. 621 and 623. It has very satisfactorily decreased initial condensation by diminishing the range of temperature in each cylinder, but is undoubtedly expensive regarding first cost, a fact which presses less heavily on large powers than on small ones. The gain is due to placing re-evaporation at earlier positions in the *total* expansion.

Entropy and Temperature-entropy Diagrams.—Let the mechanical energy given to an overshot water wheel be $w(H'_1 - H'_2)$, where H'_0 is the zero of head and energy, and w the gravity weight of the water used. Applying the analogy to heat engines, we there have the heat energy given as $\phi(\tau_1 - \tau_2)$, where τ_0 is zero of temperature and of energy, and ϕ is what Zeuner calls

heat-weight, Rankine the *thermodynamic function*, and Clausius the *entropy*; the last term being now universally applied.

We may now draw a diagram of heat changes by plotting τ to a base of ϕ as in Fig. 844, from which we may deduce

$$\text{Heat supplied or rejected} \doteq \phi \times \text{mean } \tau.$$

Just as the pv curve shews work done, and is a cycle of mechanical operations, so the $\tau\phi$ curve shews heat change, and is a cycle of thermal operations. It was first practically developed by Mr. J. McFarlane Gray, and its great value is apparent when one remembers how much heat is unrepresented on the pv diagram; though both diagrams have their uses. Isothermals, having constant temperature, are horizontal straight lines on the $\tau\phi$ diagram; and adiabatics are vertical straight lines, for no heat is being supplied and yet τ changes, $\phi\,(\tau_1 - \tau_2) = 0$ and $\phi = 0$, or entropy is unaltered (see Fig. 844).

Applying to the Carnot cycle, Fig. 844, heat supplied is the area H_1, and that rejected the area H_2, for A B, C D are isothermals, and B C, D A adiabatics. Also the work done is $H_1 - H_2$

$$\therefore \text{ Efficiency} = \frac{\text{work done}}{\text{heat expended}} = \frac{H_1 - H_2}{H_1} = \frac{\tau_1 - \tau_2}{\tau_1}$$

To draw the $\tau\phi$ diagram for water and steam, shewing heat supplied per lb. weight, we commence with an arbitrary zero of entropy at 32° Fahr. = 492° absolute. Then

$$\text{Entropy of water at } 492^\circ = 0$$

and at any other temperature τ

$$\phi_w = \log_e\tau - \log_e 492 = \log_e\tau - 6{\cdot}198\,^*$$

from which Table I. (next page) is calculated, and the results plotted as curve A B, Fig. 845, the area under which shews heat supplied up to any temperature.

* Hyperbolic log. = common log. × 2·3026.

I. Entropy of Water from 32° Fahr.

PER LB. WEIGHT.

$t°$	τ	ϕ_w
32	492	·000
50	510	·036
100	560	·129
150	610	·21[illegible]
200	660	·295
250	710	·367
300	760	·438
350	810	·504
400	860	·566

We have next to find the additional entropy due to the conver sion of water into steam. Now at any temperature τ_1 the sensible heat from 32° is the area S_h, and the heat to be supplied in making this into steam being area L_h, the entropy added $= L_h \div \tau = \phi_s$. In this manner the second curve CD has been drawn and the figures in Tables II. and III. obtained:

II. Entropy of Steam per lb. wt.

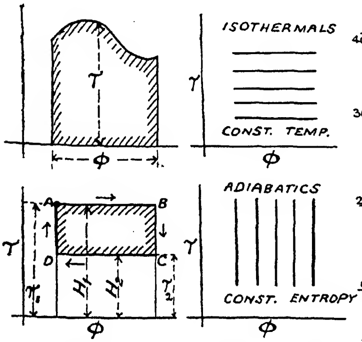
τ	$\phi_s = \frac{L}{\tau}$
492	2·219
510	2·116
560	1·865
610	1·655
660	1·476
710	1·322
760	1·188
810	1·070
860	·965

III. Entropy of Steam and Water per lb. wt.

τ	$\phi_w + \phi_s$
492	2·219
510	2·188
560	1·994
610	1·870
660	1·771
710	1·691
760	1·626
810	1·574
860	1·531

The next problem is to convert an ordinary $p\,v$ indicator n into a $\tau\phi$ diagram, which we shall here solve by the

method of Prof. Boulvin, shewn in Fig. 847. Draw four axes: temperature vertically upward to any scale, from 100° to 400°



Entropy. Fig. 844.

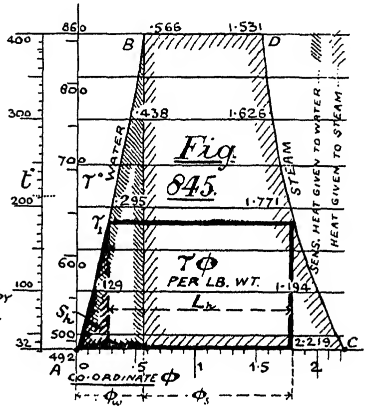


Fig. 845.

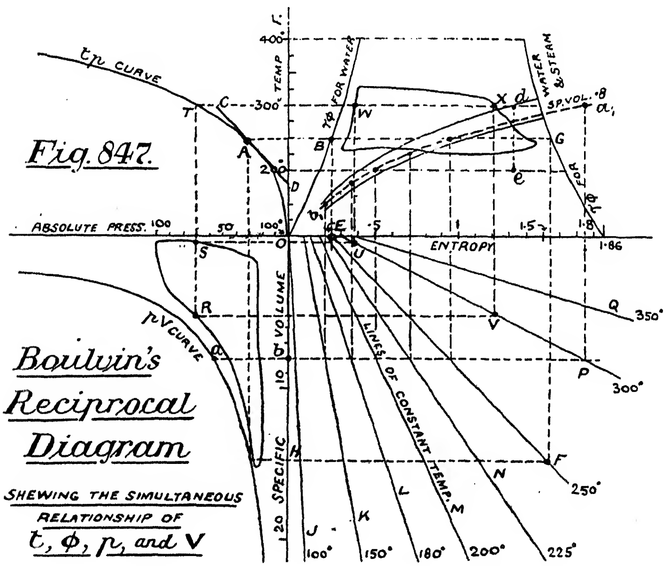


Fig. 847.

Boulvin's Reciprocal Diagram

SHEWING THE SIMULTANEOUS RELATIONSHIP OF

$t, \phi, p,$ and V

Fahr., absolute pressure leftward to any scale, entropy rightward to any scale calling ϕ at $100° = 0$, and specific volume downward by a method to be described. Construct the *tp* curve from p. 777, and $\tau\phi$ curves as already explained; then take any point A on the *tp* curve and draw a tangent C D. Project A B G horizontally and B E vertically; and make E F ∥ C D. Next project G F downward

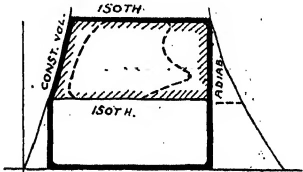


Fig. 848.

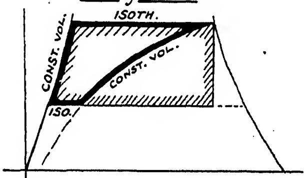


Fig. 849.

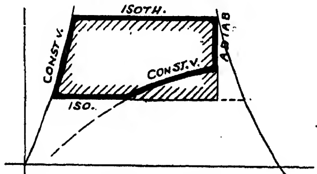


Fig. 850.

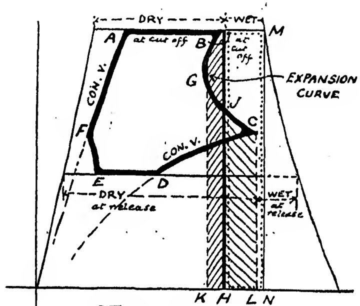


Fig. 851.

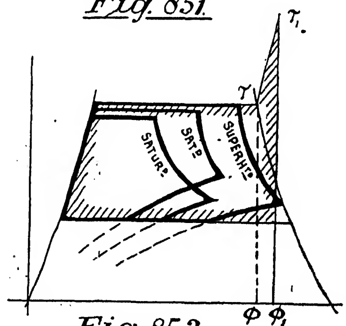


Fig. 852.

Temperature-Entropy Diagrams.

to cut E F, and draw F H horizontally. O H is now to be divided according to the specific volume at the given temperature, and thus a scale of volumes is obtained from which the saturation curve pV can be drawn. Following the construction described, the radial lines J, K, L, M, N, P, Q are next found, which are lines of constant temperature. Now take any indicator card, as at

p. 764, whose relation to the dry curve has been found, and plot it, point for point, so that it bears the same relation to pV. The transformation from $p\,v$ to $\tau\,\phi$ ordinates is next made : thus R and S are projected vertically to T, and horizontally to U and V respectively, on the proper constant-temperature line, and lastly U and V are projected vertically to W and X, giving points on the $\tau\phi$ diagram.

Note that a *constant volume* line $a\,b$ on the $p\,v$ diagram, when projected rightward and upward at various temperatures, becomes the curve $b_1\,a_1$ on the $\tau\,\phi$ diagram; and the adiabatic $d\,e$ on the $\tau\,\phi$ diagram can be similarly projected back to the $p\,v$ diagram. Constant pressure lines are always parallel to base lines.

Finally, before leaving this important heat diagram, a few further applications will be shewn. Fig. 848 is the Rankine cycle, where heat supplied is shewn by thick lines and the work done by shaded area; then, ideal efficiency is calculated by taking the base line at absolute zero. If the dotted figure be a card plotted out, its included area divided by the shaded area will be the *comparison-ratio.* Fig. 849 shews steam used non-expansively but without the usual losses. Dividing the black area by the shaded portion will give the comparison-ratio. Fig. 850 is the usual indicator card with incomplete expansion, but without cylinder condensation. Fig. 851 is a good indicator card with sharp corners, and the decrease of work done is easily seen; due to wetness from B to C, drop at release from C to D, compression at E F, and clearance at E A. As B H is an adiabatic, the area B G K H is additional liquefaction loss after cut-off, H J C L the gain by re-evaporation, and H B M N the loss before cut-off. Fig. 852 is for steam superheated before entering the cylinder, where ϕ is the entropy of the saturated steam, and ϕ_1 that of the superheated steam, the addition $\phi_1 - \phi$ being found from the formula

$$\cdot 48\ (\log_e \tau_1 - \log_e \tau)$$

There is nothing to be got by considering superheating by pure theory. The only way is to compare the indicated heat, shewn by black lines, of an actual engine, with the shaded area, when the direction of saving is shewn, being a decrease of the missing quantity when superheating.

P. 618. **The Indicator.**—Certain errors are possible in all indicators and their gears, which should be eliminated or allowed for. (1) The pressure springs should be tested by connecting the indicator to a mercury gauge, and admitting steam to both simultaneously, up to full load. A coefficient is thus deduced to correct indicator readings. (2) The drum spring being light, is sufficiently corrected by the use of wires instead of strings. (3) The decreasing gear should be mechanically accurate and rigid. (4) The connecting pipes P R Q, Fig. 621, are best dispensed with for very accurate purposes, and two indicators used, one at P and the other at Q.

P. 627. **Willans' Law** was discovered by the late P. W.

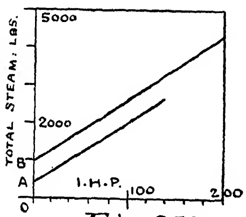


Fig. 853.

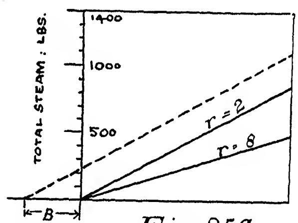


Fig. 854.

The Willans Line.

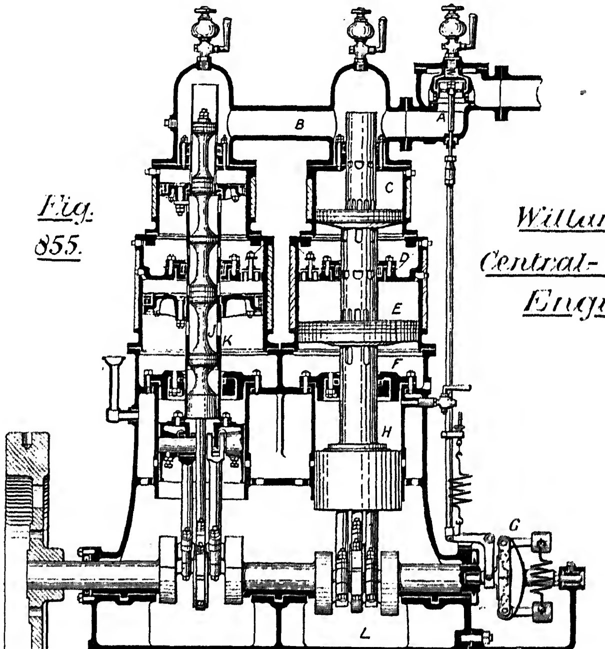
Willans when testing his engines. It occurred to him to plot total steam consumption per hour to a base of H.P. (brake, indicated, or electrical) and as the latter was varied by throttling the steam at *constant expansion*, he found the plottings to follow an inclined straight line passing above the origin, shewing total steam consumption to be proportional to H.P. plus a constant. In Fig. 853 is a pair of these lines for Willans engines, where the heights O A and O B represent steam consumption for no work, viz., condensation and leakage. Gas-engine lines pass nearly through O, but oil engines have similar losses to steam engines. Another method of plotting is to use piston mean-pressure as base, Fig. 854, and two lines are shewn for different expansion ratios in a good condensing engine, the steam consumption being reckoned theoretically as on p. 627, but neglecting factors D_f and L_q; there herefore, no loss at the origin. Similarly the dotted line is the

heoretical statement of a non-condensing engine, where B is back pressure of atmosphere.

P. 633. **High-speed Engines.**—These cannot be dismissed without mentioning two important examples.

The Willans Central-valve Engine has shewn exceptional economy. Fig. 855 is a section through a compound type, where

Fig. 855.

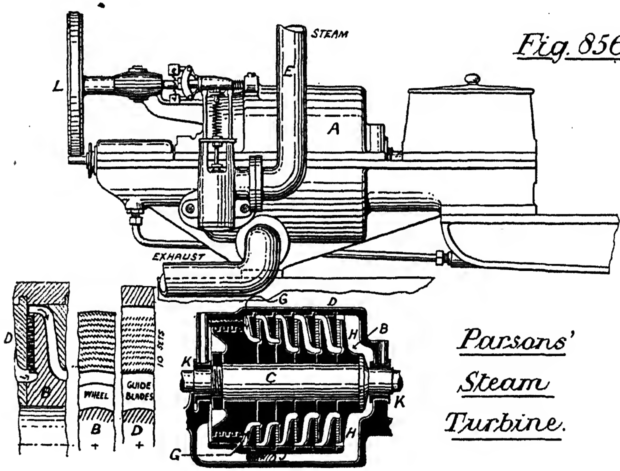


Willans' Central-valve Engine.

A is the throttle valve controlled by a governor G, B the steam chest, C the high-pressure and E the low-pressure cylinders, D the intermediate receiver, F the exhaust chamber, H the air cylinders, and J the slide valve with piston trunk K. The eccentric being on the crank pin, J and K move relatively by the amount of eccentric travel, and the cranks are 180° apart. Steam passes from B to C by the trunk, then similarly to D, E, and F; but work is only done on the down stroke, the pistons being in equilibrium

on the up stroke as in the Cornish cycle, p. 628, thus dividing the temperature drop in each cylinder on a semi-compound principle. Some air being caught in H on the up stroke, serves as a cushion compelling none but compression stresses on the crank, thus avoiding the reversal knock and unnecessary alteration of steam distribution. The piston and gland rings are all steam packed. The advantages of such single-acting high-speed engines are (1) balance due to opposite cranks, (2) economy of the Cornish cycle or 'transfer system,' (3) constant direction of stress, (4) less cylinder condensation, and jackets unnecessary, (5) more regular

Fig. 856.



Parsons' Steam Turbine.

turning effort due to high speed, (6) direct adaptation to dynamo driving, (7) lighter engine for given power.

The Parsons Steam Turbine is best introduced by a reading of the matter on water turbines in Chap. XI. The first steam turbine had parallel flow, and was virtually an impulse machine where the wheel travelled at 40 % of the steam velocity; the latter being 380 ft. per second, and the former making 9000 revs. per m. Per H.P. hour 42 lbs of steam was used, entering at 84 and leaving at 15 lbs. absolute pressure. The present motor has often outward or inward flow, and by compounding or causing the steam to pass successively through five or seven wheels, the

steam *pressure* is used rather than mere impulse, resulting in a consumption of 16 lbs. per I. H. P. hour, while running at 4800 revs. per m.; and the absolute pressures are 115 lbs. and 1 lb. when entering and leaving respectively. In Fig. 856 the casing A contains five wheels B B fastened to shaft C, and five fixed discs D D carrying the guide blades. As the steam, which is superheated, passes between B and D in each case, it is deflected by the guide blades against the wheel blades, causing rotation. Entering by pipe E, through throttle valve F, into space G, it passes between every set of blades till it emerges at H, exhausting at J. The pressure at G being high, there is a special thrust-bearing to prevent leakage, and at K K are stuffing boxes. The governor shaft L is driven by friction gearing, and a conical cam upon it depresses the throttle valve at regular intervals, thus admitting steam intermittently. The governor therefore regulates amount of opening by sliding the cam longitudinally on shaft L. (*See pp.* 966 *and* 1168.)

P. 664. **Zeuner Diagram for Meyer Valve.** — This construction has already been shewn, where opening of main

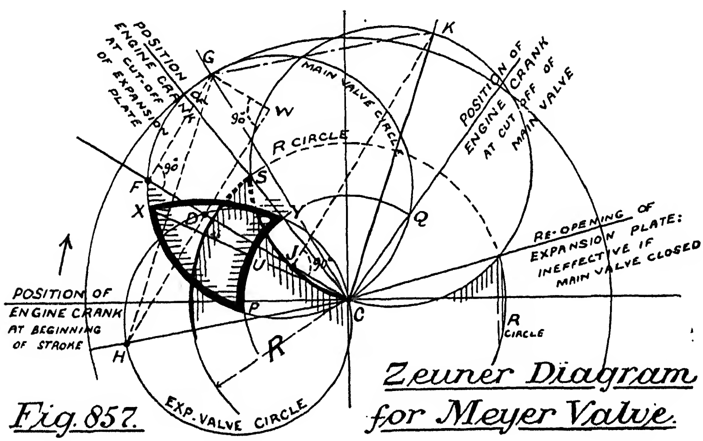


Fig. 857. Zeuner Diagram for Meyer Valve.

valve is given by the usual circle, and the closing of expansion plate by a circle of relative motion. The construction may be proved from Fig. 857, where H C is the expansion-valve circle, G C

the main-valve circle, and C K that shewing relative motion of the two valves. Join H D, and produce to W, making G W ∥ F D, and join K J. We must prove J C = F D, the difference- or relative-motion. Now F, D, J, and W are right angles (*Euc.* III. 31), and G H = K C; therefore J C = G W = F D. Hence the radii-vector of the vertical shading shews diminishing opening of expansion plate, which becomes zero when C S = the R circle radius or circle of negative steam lap of expansion plates. Also the horizontal shading, radially intercepted, gives opening of main valve. Combine the two areas by drawing radial lines through C, taking the shorter intercept of the two, and placing radially on base P Y, and thus obtain the curve P X Y, shewing that opening from P to X is governed by main valve, and closing from X to Y by expansion plate: the two spaces being equal at X U.

P. 686. **Condensers.** — Wherever the supply of cooling water is limited, the *Evaporative Condenser*, Fig. 858, may be adopted. The exhaust steam A passes through a large number of pipes F, cooled externally, and the condensed steam is returned to hot well by pump B. The upper trough C is kept filled with water, which trickles over the pipes into the lower trough D, and is then returned by circulating pump G to C, the evaporation loss being made good from the main at E. Whereas in the usual condensers the steam heat is simply used to raise the temperature of water, requiring about 20 lbs. water per lb. of steam (see p. 598), this condenser *evaporates* a large portion of the cooling water, thus causing each lb. of evaporated water to abstract the heat in 1 lb. of steam or thereabouts; hence the weight of cooling water may be merely *equal* to that of the condensed steam. In practice it is often less, for radiation and conduction do some of the work The evaporation is increased some 50% if fan draught is used; and 10 sq. ft. of pipe surface per I. H. P. is adopted, as against 1 or 2 with surface condensers. The joints must be specially good, but a 24 in. or 26 in. vacuum may be obtained, and still better results if the steam and water currents be opposed in direction. External incrustation must be removed at times.

Various cooling systems are also adopted to economise condensation water. In Fig. 859, A is an 'independent' Worthington jet condenser, where B is the steam cylinder and C the air

pump. The injection water being drawn through D, and the exhaust entering at E, the condensed steam and water is delivered through F, and distributed by the revolving pipe G. Trickling down the drain pipes at H it finally arrives at the tank J, having been cooled by the draught of fan K.

Evaporative Condenser.

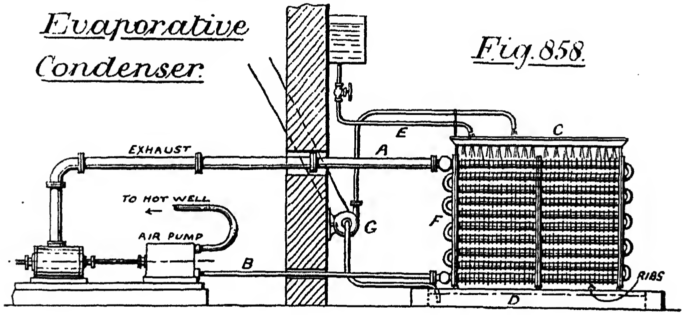
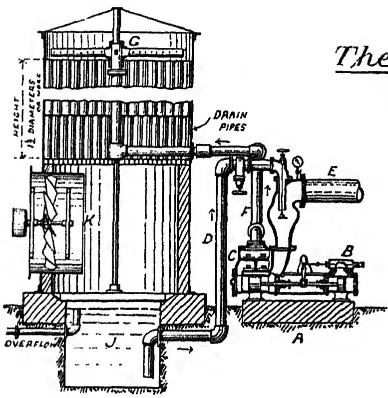


Fig. 858.



The Worthington Independent Jet Condenser

WITH COOLING TOWER

Fig. 859.

Balancing.—When an engine is perfectly balanced there will be no shock to the foundations, nor indeed are they in that case necessary; but without such balance, a heavy foundation must be provided to absorb the momenta of moving parts. Balancing may be obtained in more than one way, but the principle is always the opposition of the motion of some heavy mass to that of the engine parts, in such wise as to make the opposed momenta changes, or accelerations, nearly equal at all

times. The first difficulty is that some parts rotate while others reciprocate. Now the rotating weights are easily balanced by others having equal moment, but the reciprocating weights must be opposed by other reciprocating weights, as Mr Yarrow did in small marine engines, by means of "bob-weights" driven by eccentrics, a, Fig. 860. Large engines could scarcely be so treated, so they are still left greatly unbalanced. One of the best methods, b, Fig. 860, is that of the double-cylinder reciprocating engine with opposite cranks, which leaves only an unbalanced moment, whose arm is the distance between cylinder centres. But

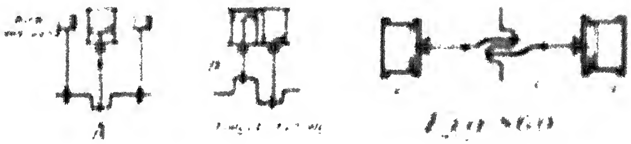
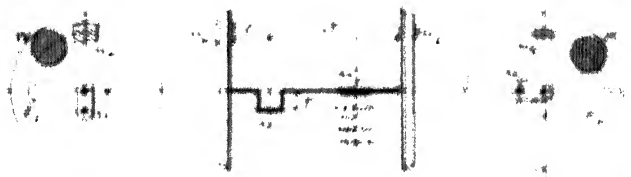


Fig. 860.



Balancing. Fig. 861.

even this can be eliminated by placing the cylinders as at c, Fig. 860.

Locomotives, unless fairly well balanced, experience dangerous oscillations, so the whole of the rotating weights, and [illegible] or [illegible] of the reciprocating weights, are considered as collected at the crank pins, where they are opposed by weights on the driving wheels. Supposing this done, as in Fig. 861, take w alone and balance it by w_a and w_b so that

$$w_a r_a + w_b r_b = w r \qquad \text{also } w_a c = w_b d$$

Next treat for w_1 similarly, finding x_a and x_b. Lastly combine w_a and x_b into one weight at their common centre of gravity, and do the same for w_b and x_a, whence W is deduced for any radius, keeping moments equal. It is customary to bolt weights W W temporarily, and drive the engine while slung in chains, so as

[illegible] the unbalanced oscillations and make corrections. It will be found that a complete balance of reciprocating parts at [illegible] produces a large unbalanced vertical shock of [illegible] *See* §§ [illegible] *and* [illegible]

[illegible] **Care of Boilers.** The possible defects in steam boilers may be thus classified:

[illegible]	[illegible]	Internal
Design	Bad Setting	Corrosion
Materials	Wasting	Grooving
Workmanship	Bad Fittings	Incrustation

and the [illegible] of such defects is emphasised in the following table:

[illegible] FOR 18½ YEARS, ENDING Dec. 1896, and showing numbers of explosions and collapses, with their causes.

Causes	Explosions	Collapses
Internal corrosion	11	22
External corrosion	55	25
Corrosion on both sides	1	1
Corrosion, side not stated	9	28
Grooving	26	1
Excessive pressure	18	18
Failure of stays	16	1
Deficiency of water	12	46
Structural weakness	11	21
Fracture	6	9
General deterioration	5	0
Seam rips	5	0
[illegible]	4	8
Not ascertained	1	4

Constructive defects occur less often nowadays, the necessity for both strength and elasticity being fully recognised, good mild steel being employed with larger plates and fewer seams, drilled holes, and hydraulically-closed rivets, and bad caulking eliminated. Many small details still need insistence, however, such as the strengthening of manholes by riveted rings, and the use of riveted bosses for all coverings so as to avoid unseen leakage. The setting should be on a dry and strong site, and should have large flues to facilitate examination.

External wasting causes many explosions, being usually con cealed by brickwork or other covering, thus retaining moisture if in a damp situation, and teaching the importance of narrow surface where brickwork meets the boiler. Smooth wasting is due to intermittent use, and is caused by moist soot, its action being so slow and uniform as to be undetected without a drilling of the plate. Wasting also occurs at the lower part of boilers, on account of moist ashes. The fittings cannot be too carefully watched and tested: there should be two water-gauges, and the safety valves be direct-loaded if possible, while the fusible-plug metal should be renewed annually.

Corrosion or pitting is caused by the chemical action of gases in the water, or by electric action, the plate becoming electro-positive to the impurities, and the water, if acid or salty, forming an electrolyte between the two. Slightly muddy water may thus be a distinct advantage, and a little lime deposit prevent chemical action. If the boiler needs to stand a long time it should be emptied and dried, while electric action is sometimes prevented by suspending zinc blocks, connected by soldered wires to the plate, which they protect by taking its place and wasting instead.

Grooving is a surface cracking due to abrupt bending under alternate heating and cooling, and is generally found near rigid stays. When first appearing it is called 'mechanical' grooving, but, as it widens and deepens by corrosion, it is termed 'corrosional' grooving, and the obvious remedy is sufficient elasticity with strength, while rounding stay edges at the plate.

Incrustation or scale is the hard deposit in boilers resulting from boiling water at high temperatures. It depends on feed-water composition, of which the following are typical analyses:—

	Spring Water	River Water	Sea Water
1 Calcium Carbonate	13.34	8.96	2.76
2 Magnesium Carbonate	1.11	1.03	Trace
3 Calcium Sulphate	4.46	2.39	77.46
4 Magnesium Sulphate	0.77	0.05	120.65
5 Sodium Chloride	2.14	1.39	1576.11
6 Magnesium Chloride	0.56	1.19	185.06
7 Silica, Alumina, &c.	0.24	0.22	Trace
Grains per gallon	22.81	17.69	1919.32

The first and second exist in solution with the gases in the water, as bicarbonates, and are precipitated as monocarbonates. Calcium *bicarbonate* is less soluble in hot water, and is therefore reduced by heating. But the best method is the Clark process, where slaked lime is added to the feed settling tank, causing a precipitate of calcium carbonate *before* entering the boiler. Only the exact amount should be added and the feed then be filtered through stretched cloths. Another method is to add soda (caustic or carbonate) in the boiler. The former is preferable, as more energetic, but attacks brass fittings, and leaks through otherwise tight joints. The resulting loose precipitate is calcium monocarbonate, and some soda carbonate in solution, easily blown off. *Magnesium bicarbonate* is best treated by the "Clarking" process, which causes precipitates of magnesium and calcium monocarbonates. *Calcium sulphate* is very troublesome, giving water what is known as permanent hardness. The lime process is useless, and soda must be applied, precipitating calcium carbonate, and putting a sodium sulphate in solution, which must be blown off. *Magnesium sulphate* occurs infrequently in fresh water, and is only troublesome with lime salts; it may also be treated with soda, preferably in the caustic form. *Sodium chloride* (common salt) needs no mention, for fresh water is now always used in marine boilers, loss being made up by water from "evaporators." *Magnesium chloride* is objectionable by giving off

free hydrochloric acid, thus causing corrosion; but it may be neutralised with lime. If the preceding precautions be not adopted, viz., applying the right chemicals as indicated by feed-water analysis, a good deal of chipping must be done to remove scale and prevent overheating. (*See Appendix V., p.* 1006.)

P. 693. **Boiler Covering.**—To reduce radiation from boilers or pipes, pasty coverings are usually applied, which are more or less effective. The following results were obtained by Mr. Laird, of St. Louis, who made analyses and heat tests of the materials indicated:—

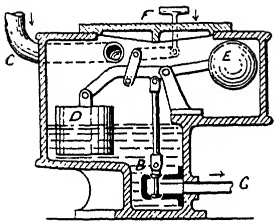
WEIGHT COMPOSITION PER CENT.

Name of Article.	Plaster of Paris.	Magnesia.	Asbestos.	Carbonaceous Matter.	Insoluble Silicates.	Soluble Mineral Matter.	Powdered Limestone.	Comparative Condensation Loss.
Plastic magnesia ...	...	93	7	...	...	...	...	334
Sectional magnesia	...	92·2	7·8	...	...	...	...	335·3
Asbestos fire-felt ...	...	82	...	18	...	...	...	367·5
Asbestos sponge, moulded	95·8	...	4·2	...	...	...	...	371·3
Fossil meal	...	...	...	12	80	8	...	376·2
Plaster of Paris and sawdust	?	...	...	?	...	...	...	438
Asbestos fire-felt cement	3·5	...	32	...	...	...	64·5	563·7
Asbestos sponge cement	10	...	31	...	...	...	59	604
Bare pipe	...	..	...	...	...	...	...	1085

P. 693. **Boiler Fittings.**—*Safety valve* area may vary considerably: thus, the Board of Trade (p. 694) require only two-thirds of the area used in America. The former, based on experiments up to 60 lbs. pressure, should be smaller at high pressures, but larger again for forced draught. The lift is always small, varying from ·2″ at 10 lbs. to ·02″ at 30 lbs. blow off, but is proportionately larger with a small than a large valve.

Steam Traps are for automatically removing condensed water in steam pipes, in one of two ways: either the rise of water lifts

at which opens a valve periodically, or the deposition of :r water contracts a plate or rod which opens the discharge :. Fig. 862 is a trap of the former type. The float D,

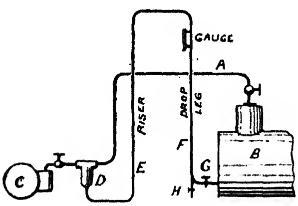


Steam Trap

Fig. 862.

ıced at E, is gradually lifted, and when it reaches a certain ht, the valve B opens, and pours the condensation water ugh G. F is a trial handle.

Γhe *Steam Loop* is an interesting contrivance by which the :r from a separator is returned to the boiler. The usual supply A, Fig. 863, connects boiler B with cylinder C, and the steam

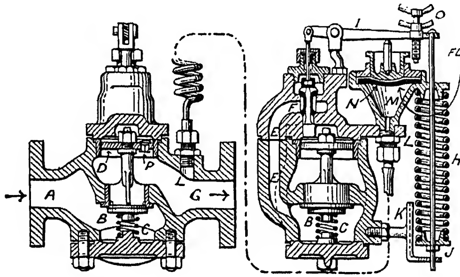


The Steam Loop.

Fig. 863.

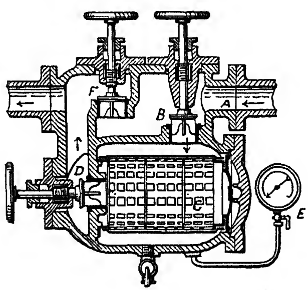
lried by separator D. From the separator drain, the water is .rned by riser E and drop-leg F, through the check valve G. v let the drain cock H be opened, so as to blow out the :s E and F: when closing again, water will accumulate in the p-leg until there is sufficient weight to open the check valve G, le the steam behind it balances the boiler pressure. In this ıner there will be an intermittent flow of water back to the .er, for the priming water will be carried up E as spray, and :n a sufficient weight has condensed in F, the check valve will ›pen.

To avoid the evils of varying boiler pressure, a *reducing valve* is now often used between boiler and cylinder, giving a lower pressure, but steadiness and dryness. Reid's valve, Fig. 864, is a



Reids Reducing Valve. Fig. 864.

good example. The steam entering at A, finds valve B closed by spring C, so passes along E, and lifting the throttle valve F, acts on the piston D and overcomes the compression of spring C, thus allowing a free flow through D to G at a lower pressure. Now the coiled pipe L is connected to vessel M, which is always full of condensation water, and the low pressure acting on diaphragm N tends to lift lever I and close valve F, thus throttling the pressure on D to the right amount. The lifting tendency is also resisted by the spring H, which can be screwed up against a scale K to give and shew the required reduction.



Rankine's Marine Filter. Fig. 865.

When the hot-well water is returned to boiler, there is danger due to the deposition of cylinder oils on the furnace crowns, causing possible overheating. *Feed-water Filters* have been therefore introduced, of which Rankine's, Fig. 865, is an example.

The water is pumped through at A and passes by inlet-valve B, through the filtering cartridge C, then by the outlet-valve D to the boilers. At first the gauge E registers boiler pressure, but as the filtering cloth on grid C becomes charged with grease, the pressure rises, and, when some 20 lbs. higher than boiler pressure, valves B and D are closed and bye-pass F opened, permitting the removal of the grid to apply a new cloth.

Feed-water Heaters are contrivances for saving waste heat, by giving it to the feed water on its way to the boiler. There are three ways of doing this: (1) by intercepting some of the heat of the furnace gases when leaving the boiler, (2) by letting the feed pass through a vessel jacketed with exhaust steam, and (3) by a similar use of live steam. The first apparatus, called an

Green's Economiser.

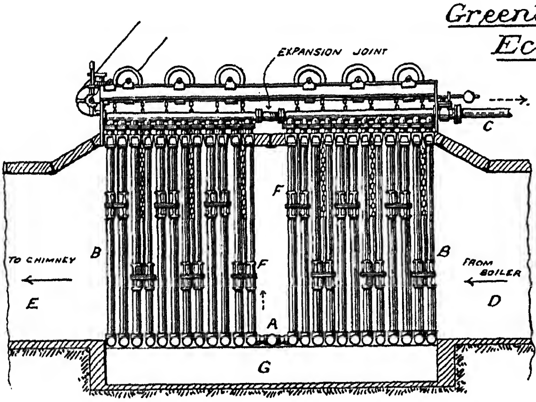


Fig. 866.

economiser,' is shewn in Fig. 866 standing in the flue between boiler and chimney. The feed-water enters at A, rises simultaneously in pipes B B, and leaves at C on its way to the boiler feed-valve, while the hot gases pass in the contrary direction, D to E. There being a large accumulation of soot, the scrapers F F, Mr. Green's invention, are used from time to time, and the débris is dropped into chamber G. Fig. 867 is an *Exhaust-steam Feed-heater*, the water entering at A and passing through the tubes, then leaving at B for the boiler; while simultaneously the exhaust steam travels by C to D, surrounding the tubes. Live-steam feed-heaters have no advantage in theory, but in practice they save

the boiler by avoiding unequal expansion due to cold feed, and also promote circulation. Injectors belong to this class.

The *Superheater*, as a boiler apparatus, has already been explained.

P. 696. **Fuel.** — The calorific value of a given fuel can be obtained as on p. 697, or may be expressed as a simple formula, where the weights of other elements are given in terms of the carbon. Thus (p. 697, lines 7 and 8) 1 lb. of H produces 4·28 times the heat 1 lb. of C does; and again (p. 697, line 20) 1 lb. of H must be deducted for every $\frac{1}{8}$ lb. of O present.

$$\therefore \left.\begin{matrix}\text{Calorific value in}\\ \text{B.T.U. per lb.}\end{matrix}\right\} = 14500 \left\{C + 4{\cdot}28\left(H - \tfrac{1}{8}O\right)\right\}$$

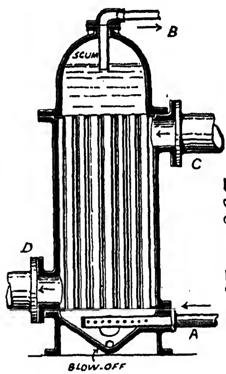


Fig. 867.

Exhaust-steam Heater.

Fuel Calorimeter.

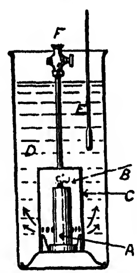


Fig. 868.

where C, H, and O are the *actual weights of these elements per lb. of fuel.* Now as 966 B.T.U. evaporate 1 lb. of water from and at 212°,

$$\text{lbs. of water evaporated per lb. fuel} = \frac{\text{calorific value}}{966}$$

given in what are called evaporation units. For the sample on p. 697 this would be 15·3 lbs. of water.

A directly practical test is, however, always advisable, and for this we may refer to Thompson's fuel calorimeter, Fig. 868. A fuel chamber A is placed on a stand, immersed in a vessel of

water B, and covered by a combustion chamber C having holes for escape of gas. E is a thermometer, and F a cock for the first rush of gas when testing, but otherwise shut. The method of use is to weigh a small quantity of dry powdered fuel, and with it the necessary amounts of potassic chlorate and potassium nitrate to produce complete combustion. These are next placed in vessel A, the fuse lighted, covered with vessel C, and the whole immersed in a known weight of water B. The fuel gradually burning, heat is given to the water, whose temperature is raised, and it is then a simple matter to find the B.T.U. per lb. of fuel. (*See p.* 1148.)

Liquid Fuel, commonly the residue 'astatki' due to the distillation of lighter oils from petroleum, is more used than formerly. The method now adopted is to induce and inject the oil as spray, by means of a steam jet, and the system has many advantages, such as very complete combustion, absence of stoking, one-third less chimney-heat loss compared with coal and twice the evaporative value, as well as a further increased evaporation for the same grate area through decreased dilution of the gases. A good 'sprayer' should give fine spray, little noise, and be quickly separable for cleaning, which requirements are well met (according to Mr. R. Wallis in his N.E. Coast Inst. paper, from which these notes are taken) by the Rusden and Eeles apparatus, Fig. 869. Both oil and steam jets are annular and regulated by hand wheels, and the complete installation, Fig. 870, consists of a pump A to raise the oil to a service tank B, thence to the sprayers, a steam pipe C for the oil feed-pump A, and another D for the sprayers, the steam being superheated on its way to secure higher efficiency and economy. The furnaces are partially lined with firebrick to distribute the heat, and to retain it for some time after the jet is extinguished. Before lighting, the furnace should be blown out by steam, and the torch applied *before* turning on the spray, otherwise an explosion is possible.

P. 698. **Artificial Draught.**—*Forced Draught* has already been explained in principle. It is practised in three different ways. The closed stokehold, with air from fans under a water head of $\frac{3}{4}''$, and air-locks for the passage of stokers, is still in operation with economical results, and without apparent injury to

Rusden & Eeles'. Liquid-fuel Sprayer.

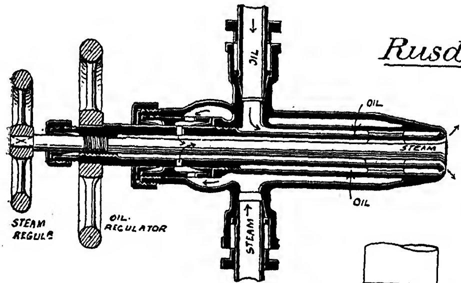
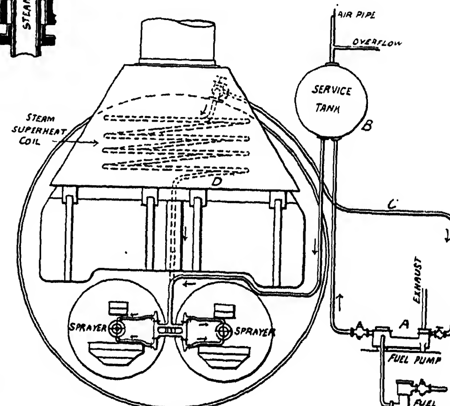


Fig. 869.



Liquid-fuel Firing Installation

Fig. 870.

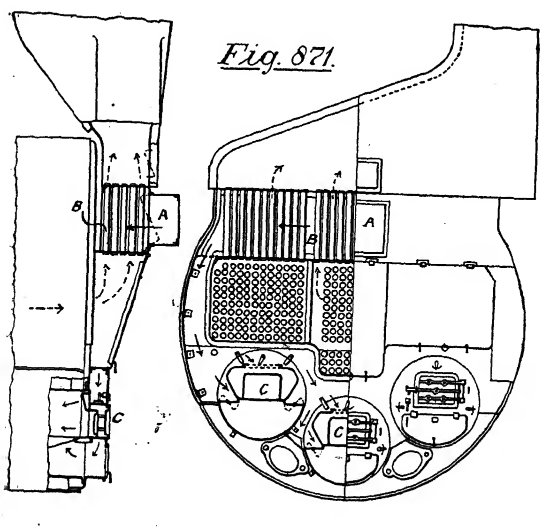


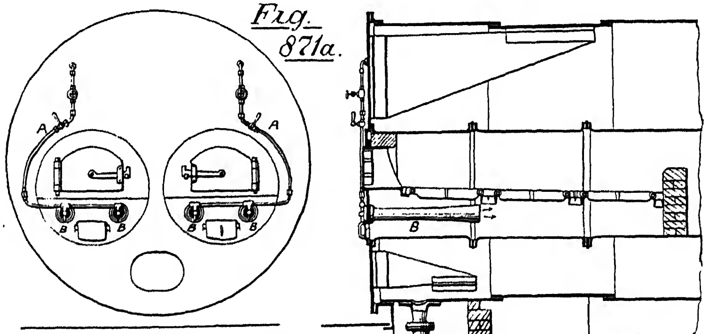
Fig. 871.

Forced Draught.

(HOWDEN'S SYSTEM)

the boiler; but it is dirty, and the stokehold temperature is as high as 116° F. The closed ashpit is shewn by the Howden system, Fig. 871, where the air is forced in at A by a fan, and passing over tubes B B that are heated by hot gases from the boiler, enters the fire through the fire-door C, both above and below the firebars. The results are more economical than in the last system, but the temperature of the stokehold is still high. In the third method, shewn by the Meldrum furnace, Fig. 871*a*, air is introduced at the blowers B, by means of a jet of steam from the pipe A, and the ashpit is closed, but the ordinary firing arrangements are not interfered with. The system is especially suitable for dust fuels.

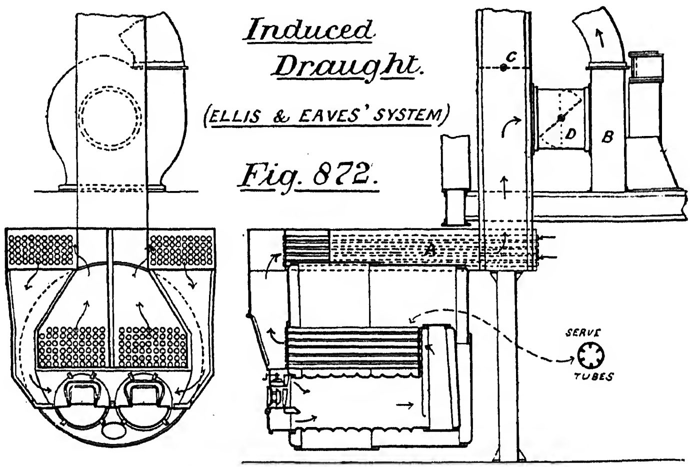
Fig. 871a.



Forced Draught. MELDRUM'S SYSTEM.

Induced draught has many advocates. It is effected by a fan in the uptake, which removes air from the boiler tubes and causes a partial vacuum into which the combustion air enters by passing over the fire. Its best representative is the Ellis and Eaves system, Fig. 872. The air enters by tubes A, where it is heated by furnace gases, and then through the ashpit door as before. Upon reaching the uptake it is deflected through the suction fan B, or may ascend directly to the chimney if damper C be opened and D closed. Its efficiency is greatly increased by the Serve tubes shewn, and retarding plates, and the stokehold is both clean and cool. There is said to be decreased injury to boiler tubes through the air entering at their centres instead of impinging on their edges, and an evaporation of 60 lbs. per sq. ft. of grate is easily obtained.

Locomotive draught is induced up to 10″ of water-head, and forced up to 2″ by the current entering the front of ashpan.



Induced Draught.
(ELLIS & EAVES' SYSTEM)
Fig. 872.

P. 698. **Mechanical Stoking.**—It is well known that hand firing is responsible for a large quantity of imperfectly-burned fuel, as well as the emission of much smoke, and some form of machine stoking is therefore desirable, whose expected advantages would be (1) smoke-prevention by uniform firing and constantly closed doors, (2) economy of fuel by using cheap coal and obtaining more perfect combustion, (3) economy of labour, (4) higher, more uniform, and more easily regulated evaporation. There are two forms of these machines : *coking stokers*, where the coal is first fed to a dead-plate or its equivalent, for preliminary distillation ; and *sprinklers*, where the fine coal enters at the centre of the bars and is distributed by fans or beaters. We have only space for an example of the first, Fig. 873, as made by the New Conveyor Company. The hoppers A A contain small coal, placed there by mechanical elevators. This coal is fed slowly forward by the screw B, then coked at C, and pushed forward gradually by the movement of the firebars, which are connected to the crank shaft D. Finally the fuel arrives at E completely burnt, and simply drops down as ash.

P. 699. **The Gas Engine.**—Beau de Rochas shewed the necessity for large cylinder volume and small circumference, high piston speed, large expansion, and high initial pressure, proposing

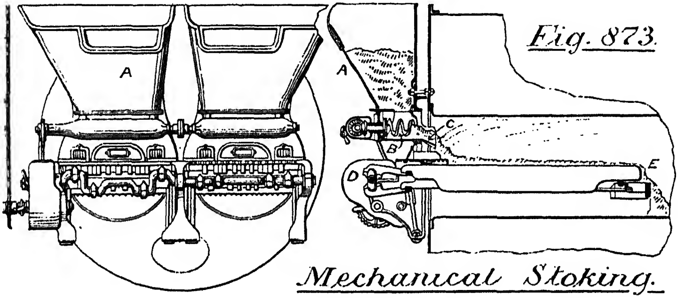
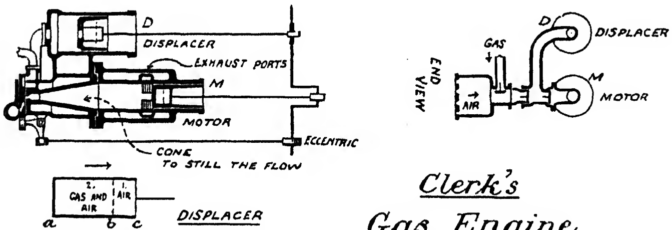


Fig. 873.

Mechanical Stoking.

also his four-stroke cycle, where he used compression to give opportunity for sustained expansion. The defects of this cycle are irregular crank effort, dilution of charge with burnt gases, and expansion-ratio no larger than compression-ratio.

Other Cycles.—In Clerk's engine of 1880, Fig. 874, the charge



Clerk's Gas Engine. Fig. 874.

is first prepared in the displacer cylinder whose strokes are:

→ admission of charge, first gas and air, afterwards air.

← slight compression, and expulsion to motor.

The motor crank follows that of displacer by 90°, and the motor-cylinder operations are

End of outward stroke Exhaust and [illegible] admission of [illegible] charge, [illegible]

" " Compression of charge

" " Ignition, expansion, and exhaust, as before

A slight loss of charge occurs when scavenging, but higher ignition pressure is obtained, and [illegible] per [illegible]. The Beck Engine, 1888, used a six-stroke cycle, two strokes of which were for scavenging by the admission and expulsion respectively of pure air, but crank effort was very uneven. In the Griffin Engine the same arrangement is adopted, but the

Atkinson's Gas Engine

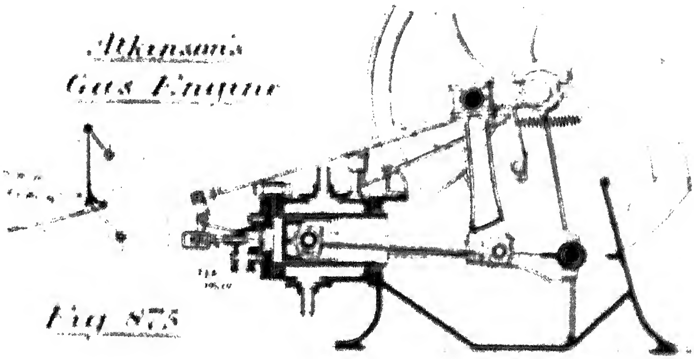


Fig. 875

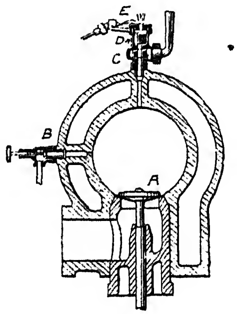
piston being double-acting there is an impulse every three strokes. Undue heating, the chief objection, is removable by the adoption of two single-acting cylinders. Two ingenious engines made by Atkinson used a link-work driving-gear to so vary the piston speed as to give long expansion and short compression strokes. Thus in a 2 H.P. (nominal) 'Cycle' engine, the second form, the following proportions obtained :—

Length of admission stroke	6·3"	11·33"	about 1 : 2
Length of compression stroke	5·03"		
Length of expansion stroke	11·13"	23·56"	
Length of exhaust stroke ...	12·43"		

and while the compression ratio was 2·6, that of expansion was

4·3. Fig. 875 shews the engine, whose only objection was complication and weight of parts.

Self-starter.—There are three methods of starting a gas engine: (1) by turning round during four strokes, igniting, and keeping moving for a short time, (2) by the use of a small turning engine for large examples, (3) by attaching a 'self-starter' to carry the piston through four strokes. In Lanchester's apparatus, Fig. 876, the exhaust-valve A is held up during expansion and compression by a special cam, which slips out of gear after starting; while B is a cock for admitting gas, and C is open for the escape of gas and air, the mixture being lit by the flame E. Having placed the crank past dead-centre, the flame C is watched: at first it is blue, then brighter, and just as it begins



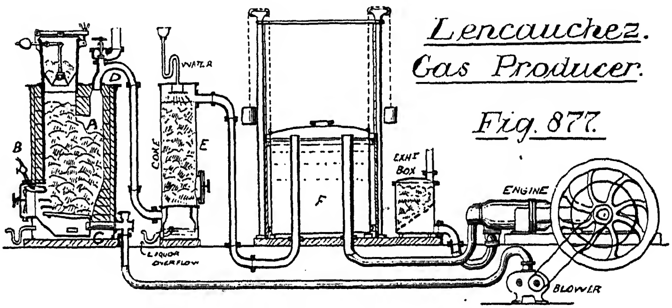
Lanchester's Self-Starter FOR GAS ENGINES.

Fig. 876.

to roar, B is closed, which causes the flame to strike back into the cylinder and explode the charge, non-return valve D lifting automatically to prevent escape. This impulse should carry the piston through four strokes, and very little aid continue the motion.

For engines of 8 or 10 H.P. gas is most conveniently obtained from company's mains, but for larger engines a special plant for cheap gas is advisable. There are two methods of procuring gas from solid fuel: (1) by *distillation* in retorts and subsequent purification, (2) by combustion. The latter may be practised in two ways. *Producer gas* is obtained by burning coke to form CO, the air supply being restricted. *Water gas* is made by burning coke to incandescence and then directing a jet of steam

upon it: the latter being decomposed into H and O recombines with the coke and forms a rich gas; but as continuous production is impossible, there is a 'blow' for ten minutes, and then a re-admission of air to revive combustion. *Power gas,* known as Dowson gas in England and Lencauchez gas in France, is made from coke or anthracite (so as to avoid tar), and can be worked continuously, the steam and air being admitted together in proper proportions. Fig. 877 shews a Buire-Lencauchez gas plant,



Lencauchez. Gas Producer.

Fig. 877.

where A is the coke furnace, supplied with fuel at the top, with steam at B, and with air at C. The gas escapes at D, and passes through the 'scrubber' E, which is filled with coke to spread the trickling water and so absorb ammonia; thence to holder F, from which it is drawn by the engine. One volume of gas to $2\frac{1}{2}$ of air is required, so an engine designed for lighting-gas must have its valves altered; a scavenging charge is also necessary. The following consumptions were obtained with lighting gas:—

CUBIC FEET OF GAS USED PER HOUR.

	Lenoir.	Hugon.	Otto-Langen.	Otto-Crossley.	Atkinson.	Beck.	Griffin.
Per I.H.P. hr.	88 to 105	77 to 92	26	17 to 25	19·22	21·5	
Per B.H.P. hr.	...	...	...	19·7 to 29	22·5	26·8	23 to 28

(*See p.* 1152.)

and the next figures compare working cost of various power-producers.

COMPARATIVE COST OF ONE I.H.P. HR.

Steam	$\frac{1}{4}d.$	Petroleum	$\frac{1}{2}d.$
Gas (Lighting) .	$\frac{1}{2}d.$ to $\frac{3}{4}d.$	Electricity (from main) ...	$2d.$
Gas (Poor) ...	$\frac{1}{4}d.$ or less.	Hydraulics (from main) ...	$2\frac{1}{4}d.$

P. 709. **Oil Engines.**—For motor-cars the Priestman

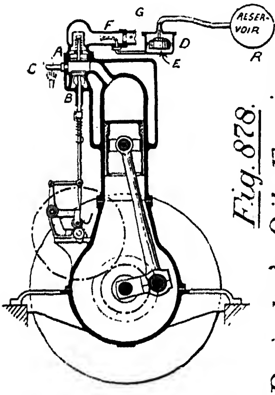


Fig. 878. Daimler's Oil-Engine. FOR MOTOR CARS.

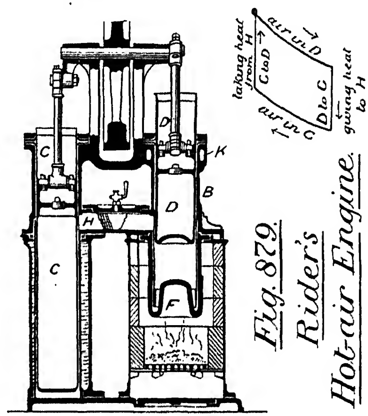


Fig. 879. Rider's Hot-air Engine.

engine was too heavy, so Daimler and others designed lighter forms, Fig. 878. There were two cylinders, side by side, having cranks at 180°, and using the Otto cycle, so there was an explosion per revolution. The charge of oil-vapour and air entered at A, the exhaust leaving at B; and the ignition tube C, of platinum, was kept hot with a lamp. The oil, benzoline, flowed from reservoir R through the chamber D, at a slow and regular rate, controlled by float E, and as it emerged at F was wafted into spray by the suction air at G. The exhaust-valve B was lifted every second revolution by a cam J, worked from crank shaft by gearing. Further information on motor-cars and their details is given in the Appendices at pp. 947, 965, 999, and 1182.

Hot-air Engines.—These are sufficiently illustrated by the

Rider Engine, Fig. 879, which has been much used for small powers. C is a displacer, and D the power plunger, their cranks being about 90° apart. At F is a fire (hot body), at E a water jacket (refrigerator), and H is the regenerator; the cycle being essentially that of Stirling, and the operations as follows:—

1. Air compressed in D takes heat and expands: thus work is done *by* the gas.
2. The air is transferred to C nearly at constant volume, giving heat to H.
3. Air under C is compressed, losing heat to E: thus work is done *on* the gas.
4. The air is re-transferred to D at nearly constant volume, taking heat from H.

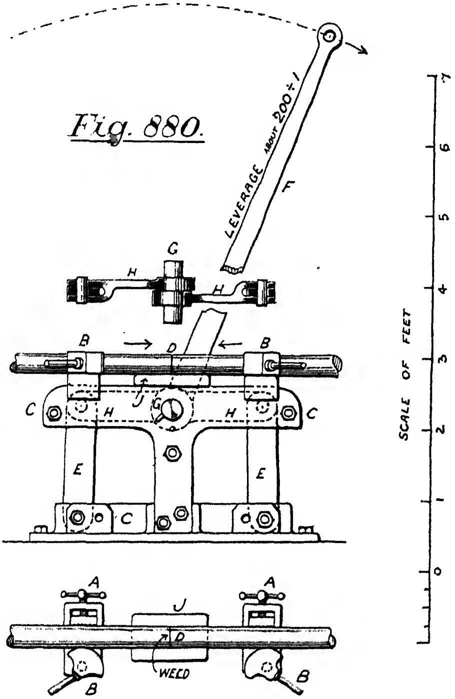
APPENDIX III.

(FOURTH EDITION.)

CHAPTER IV.

P. 102. **Welding.** — A very satisfactory and interesting welding machine, which will also serve for a certain amount of forging, has been introduced by the Nicholson Tool Company of Newcastle-on-Tyne: it is here illustrated in Fig. 880. A frame

Fig. 880.



Nicholson's Welding & Forging Machine.

C C carries a table or anvil J to support the work, and two vices B B to hold it. The latter are fixed to the upper end of the rocking levers E E, while a hand lever F operates an eccentric shaft

connected to E E by the coupling rods H H. The bars to be welded are first placed cold in the vices, and the screws A A are adjusted to suit the thickness of the work, so that the final grip may be given by the eccentric jaws B B. The welding heat is next obtained, and, while two men put the pieces quickly in the vice jaws with the hot surfaces in contact at D, a third man pulls over the lever F, and thus squeezes the joint to a perfect weld which may be finished with hammer in the usual way. The leverage is about 200 to 1, and a pressure of about 20 tons is produced at J. The machine may be further applied to jumping or upsetting, and a reversal of pull will elongate or draw out a bar

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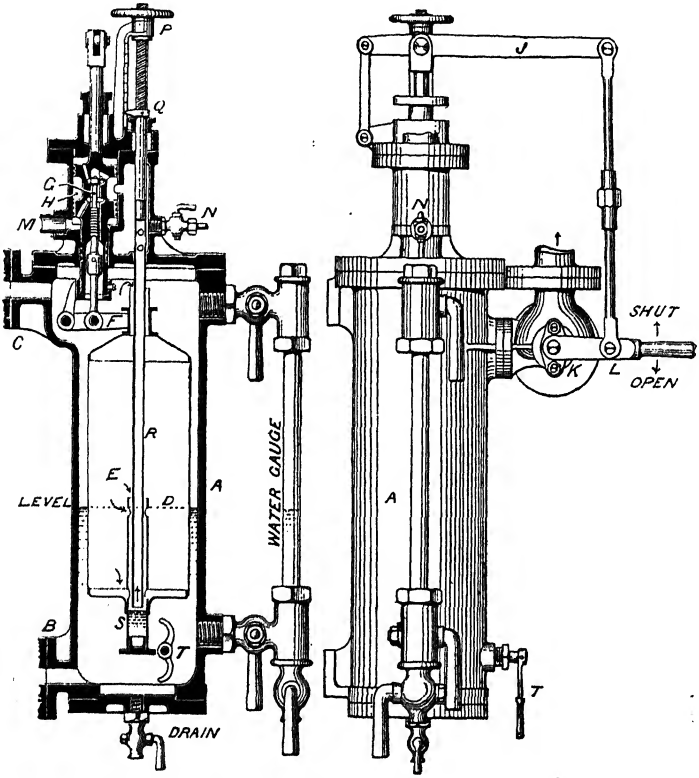
CHAPTER V.

Pp. 138 & 169. **Loss on Return Stroke.**—It is shewn in this chapter that the return stroke in reciprocating machine tools, when no useful work is done, generally takes place at a higher speed. The introduction of electric motors for individual driving has enabled us to measure the work absorbed on both advance and return strokes, and has shewn that in planing machines the total work used on each stroke is almost identical, though that of the return stroke is slightly less. This means that the work at the tool point is but a small part of the *total work* of driving and cutting. Further, if the return stroke be done in one *n*th of the time of the advance, the *horse-power* on the return is about *n* times that of the advance stroke, which, of course, follows from the previous statements. It is evident, therefore, that the quick return stroke does not save in work done per stroke, but is a saving in time, while absorbing a higher horse-power, proportional to speed.

CHAPTER VII.

P. 831. Appendix II.:—**Automatic Feed-Water Regulator.**—This apparatus is an important adjunct to any water-tube boiler, for where little water is contained the level fluctuates considerably with variable engine-power. The well-known Belleville

regulator has been much improved in Mr. Andrew Forster's invention Fig. 881. The gun-metal casing A is fixed to or near the boiler so that flange B connects to water, and C to steam. The level is



Forster's Automatic Feed Regulator. Fig. 881.

at D, but may be adjusted within certain limits. The hollow float E is connected to lever F, so as to act on the 'pilot' valve G, thus distributing steam to the large piston H, which again acts on the feed cock K through the lever J. If the feed be too slow, the

float falls, valve G moves downward, admitting steam to top of H, which thus opens feed valve still wider. But the movement of H downward also cuts off steam supply at G, and all is quiet again with increased rate of feed. Conversely, if the feed be too rapid the float rises, lifting G; and, the top of H being thus opened to exhaust M, the unbalanced steam pressure at bottom causes H to rise and slightly close the feed valve. Also the act of rising once more cuts off G and restores equilibrium with a decreased rate of feed.

To raise the water level permanently, shut off steam at C and open the blow-out cock N, thus flooding the float with water. Next, turn the screw P till pointer Q shows level required. Keeping N still open, re-open the cock at C, and the water level falls, the extra water in the float being blown out, until the steam freely flows through bottom end of pipe R, when the definite weight of water remaining at S will keep the float at the exact level indicated by Q. In lowering the water level the flooding need not be done, for when R is depressed to the proper position, the superfluous water may be removed at S by blowing through. The hand gear at T is for testing the freedom of the float. (*See p.* 1063.)

CHAPTER VIII

P. 361. **Stress.**—It is well to warn students of the various uses to which this word has been put. Most authorities agree in applying it generally to the state of the molecules; but while some also take it to mean *unit stress* or stress per square inch, speaking of total stress as *load*, others have used it for total stress only, calling unit stress *intensity of stress.* It has therefore been deemed advisable in this book to firstly use the word for *state of stress* in a general sense, and afterwards to speak more particularly of **stress per square inch**, and of **total stress.**

P. 407. **Value of w''**, or the space between rivets in a single-riveted lap joint, as deduced from the formula at head of p. 407, depends on the material of the rivet and plate. Assuming safe stresses as follows:

		steel plates	iron plates	copper plates
Tensile stress f_t	......	6 tons	4 tons	2 tons
		steel rivets	iron rivets	copper rivets
Shear stress f_s	......	5 tons	$3\frac{1}{2}$ tons	$1\frac{1}{2}$ tons

and deducing the formula for w'' as :

$$w'' = \frac{f_s}{f_t} \times \frac{3{\cdot}1416}{4 \times {\cdot}6} = 1{\cdot}31 \frac{f_s}{f_t}$$

For steel plates and steel rivets :

$$w'' = 1{\cdot}31 \times \frac{5}{6} = \underline{1{\cdot}09 \text{ ins.}}$$

For iron plates and iron rivets :

$$w'' = 1{\cdot}31 \times \frac{3{\cdot}5}{4} = \underline{1{\cdot}14 \text{ ins.}}$$

For steel plates and iron rivets :

$$w'' = 1{\cdot}31 \times \frac{3{\cdot}5}{6} = \underline{{\cdot}76 \text{ in.}}$$

For copper plates and copper rivets :

$$w'' = 1{\cdot}31 \times \frac{1{\cdot}5}{2} = \underline{{\cdot}98 \text{ in.}}$$

P. 419. Strength of Hollow Shafts.

Example 66.—A hollow steel shaft is to be designed so as to meet safely a twisting moment of 280 ton inches. The internal diameter being 3 ins., find the external diameter :

$$\frac{D^4 - d^4}{D} = \frac{16\ T_m}{f\pi} \qquad D^4 - \frac{16\ T_m}{f\pi} D = d^4$$

$$\therefore\ D^4 - \frac{16 \times 280 \times 7}{7 \times 22} D = 81 \qquad \text{or } D^4 - 203\,D - 81 = 0$$

This equation may be put in the form

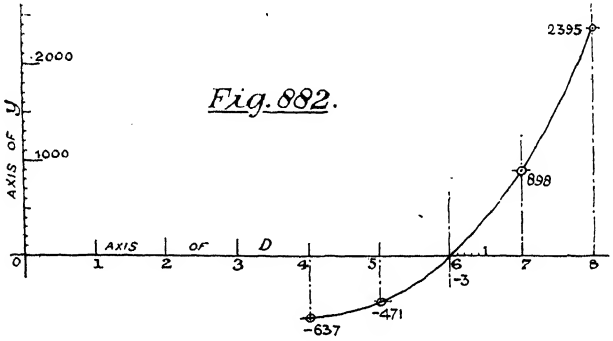
$$D^4 - 203\ D - 81 = y$$

and a curve may be constructed shewing y for various values of D, the correct value of D being that corresponding to $y = 0$. Thus :

$$\begin{array}{lll} D = 4 & y = 256 - 812 - 81 & = -637 \\ D = 5 & y = 625 - 1015 - 81 & = -471 \\ D = 6 & y = 1296 - 1218 - 81 & = -3 \\ D = 7 & y = 2400 - 1421 - 81 & = +898 \\ D = 8 & y = 4100 - 1624 - 81 & = +2395 \end{array}$$

These values of y and D have been co-ordinated in Fig. 882, and a curve has been drawn whose vertical ordinates vary from -637 to $+2395$. The point where the curve crosses the axis of D shews $D = 6$ when $y = 0$. The true answer is

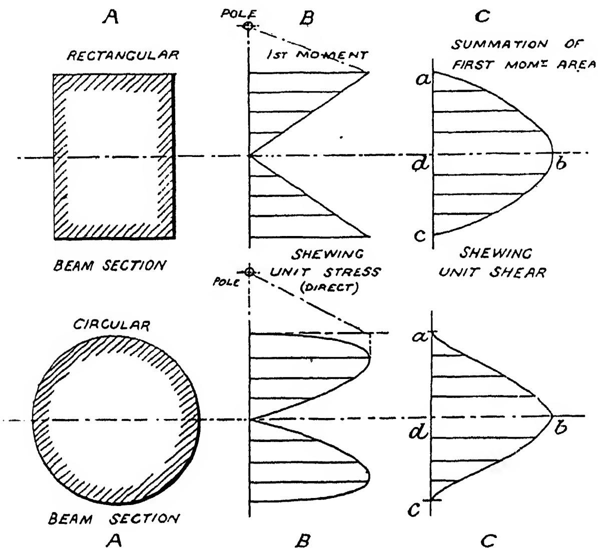
$$D = 6{\cdot}005 \text{ ins.}$$



Solving Hollow-Shaft Equation.

P. 438. **Distribution of Shear stress in Beams.**—Just as the summation of vertical forces (loads and reactions) in a beam will give a curve of total vertical shear at every section (p. 853), so the summation of horizontal forces (tensile and compressive stresses) will produce a curve of horizontal shear stress intensity; and, as the tensile stresses oppose the compressive stresses, the greatest shear is at the neutral axis. The summation (*see* p. 851) of the stress areas of p. 431 must then proceed from the upper or lower fibre to a maximum at the centre; and two examples have been given in Fig. 883 to shew how this should be done. Again, any elemental cube must be subject to vertical shear on its vertical sides, and horizontal shear on its horizontal sides, the one stress being caused by the presence of the other; hence the distribution of vertical and horizontal shear stresses over a section are identical. Finally the scale of diagrams c c, Fig. 882, can easily be found by equating the area *a b c d* to the total vertical shear at the given section; and *d b*, the maximum unit stress, is thereby arrived at.

P. 469. **Redundant Members.** — It has already been mentioned that the static force diagram shews no stress in a



Distribution of Shear Stress.
Fig. 883

redundant or superfluous member; but as soon, however, as the structure is subjected to deformation, and not till then, such a member receives stress on account of the strain in the remaining bars, the static stresses in the latter being at the same time more or less altered. The problem, therefore, can only be solved by reference to the *work done* on the bars. Now when there are no redundant members in a structure

$$\text{no. of members} = 2\ (\text{no. of joints}) - 3$$

a rule easily verified, and useful for discovering the number of extra bars.

Having ascertained these, there are two methods of finding the stresses set up in them. Assuming the whole structure to be

strained by the load, the length and cross section of every bar being known, the total work done can be expressed. When this has the lowest possible value, the true stresses in the extra bars will have been found: hence the expression is differentiated with regard to each 'redundant' stress, and equated to zero. The other method, the one which we shall here explain, is best understood by an examination of two cases.

Case I.—Imagine a properly closed structure, without extra bars, consisting, say, of four triangles. No load is applied from without, and there is therefore no stress in any member. Now let there be two superfluous members, which, being simply fitted to the framing, are not under stress: neither are the other bars stressed. Next, screw up the extra bars so as to put a stress (compressive or tensile) within them, and immediately stresses are felt upon all the other members. The work done on the extra bars will be of opposite sign to that done by them individually upon the remaining members, for, while one action tends to compress, the other tends to expand the framing. Also these two quantities must be equal, for one is caused by the other.

Let the extra bars be called *a* and *b* respectively, and the remaining bars 1, 2, 3, 4, 5 and 6. Also let

F_a	=	total stress in bar *a*
F_b	=	,, ,, ,, ,, *b*
$F_a c_{1a}$	=	,, ,, ,, ,, 1 caused by F_a
$F_b c_{2b}$	=	,, ,, ,, ,, 2 ,, ,, F_b

Then, taking the action of each force F_a and F_b separately,

work done on bar *a* = work done on other bars
(minus) (plus)

$$-\frac{F_a}{2}\Delta_a = \frac{F_a}{2}(c_{1a}\,\Delta_1 + c_{2a}\,\Delta_2 \ldots\ldots + c_{6a}\,\Delta_6) \quad \ldots\ldots\ldots(1)$$

$$-\frac{F_b}{2}\Delta_b = \frac{F_b}{2}(c_{1b}\,\Delta_1 + c_{2b}\,\Delta_2 \ldots\ldots + c_{6b}\,\Delta_6) \quad \ldots\ldots\ldots(2)$$

Clearly, then, there would be just as many equations as there are extra bars.

Case II.—Taking the same structure, let external forces be applied to it. Firstly, treat the whole figure statically, leaving

out the extra bars, and find the forces in every member, calling them F_{s1}, F_{s2}, &c. Then, taking in the extra bars,

$$\left.\begin{array}{c}\text{whole stress}\\ \text{in}\\ \text{any member}\end{array}\right\} = \text{static force} + \left\{\begin{array}{c}\text{force caused}\\ \text{by stress in}\\ \text{extra bar } a\end{array}\right\} + \left\{\begin{array}{c}\text{force caused}\\ \text{by stress in}\\ \text{extra bar } b\end{array}\right.$$

$$\therefore \quad F_1 = F_{s1} + c_{1a} F_a + c_{1b} F_b \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$
$$F_2 = F_{s2} + c_{2a} F_a + c_{2b} F_b \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

and so on for six equations, corresponding to the bars of the structure. But

$$\Delta = F\frac{l}{aE} = Fm, \text{ say} \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

Hence from (1) we have, eliminating $\frac{F_a}{2}$

$$-F_a m_a = c_{1a}F_1 m_1 + c_{2a}F_2 m_2 \ldots\ldots\ldots + c_{6a}F_6 m_6$$
(from 3 and 4)
$$= c_{1a}m_1 (F_{s1} + c_{1a}F_a + c_{1b}F_b)$$
$$+ c_{2a}m_2 (F_{s2} + c_{2a}F_a + c_{2b}F_b)$$
$$+ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$
$$+ c_{6a}m_6 (F_{s6} + c_{6a}F_a + c_{6b}F_b) \quad \ldots\ldots\ldots\ \ldots\ldots (6)$$

This is the final equation for bar *a*. From (2) we have:

$$-F_b m_b = c_{1b}m_1 (F_{s1} + c_{1a}F_a + c_{1b}F_b)$$
$$+ c_{2b}m_2 (F_{s2} + c_{2a}F_a + c_{2b}F_b)$$
$$+ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$
$$+ c_{6b}m_6 (F_{s6} + c_{6a}F_a + c_{6b}F_b) \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

which is the final equation for bar *b*.

In like manner there would always be as many equations as extra bars, and as many parts on the right side of each equation as there might be 'static' bars.

Now, in using these equations practically, F_a and F_b are not known, but $c_{1a}\ldots\ldots c_{6a}$, and $c_{1b}\ldots\ldots c_{6b}$ are known from the shape of the structure, $F_{s1}\ldots\ldots F_{s6}$ from the static diagram, and m_1 to m_6, m_a, and m_b from the length, sectional area, and E of the respective bars. Hence we have two simple simultaneous equations (6) and (7), from which to find the values F_a and F_b. When these are worked out, the stress in the other bars can be found from (3) and (4).

CHAPTER IX.

P. 510. **Proof that normal to contact passes through meet of pitch lines.**—In Fig. 884, let A and B be centres of rotation, and C the common normal to tooth surface, —— a continuous straight line because the surfaces are tangential. Also let C and D be centres of curvature.

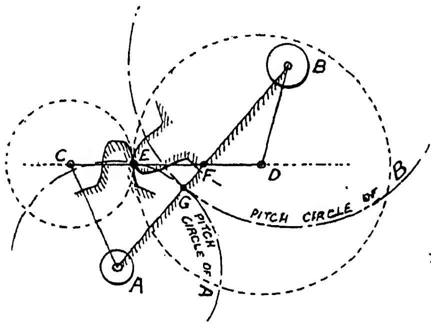


Fig. 884.

Proof OF NORMAL PASSING THROUGH MEET OF PITCH CIRCLES.

Now, for a very small motion we may substitute a quadric chain A C B D for the wheels, where A C and B D are cranks, and C D a coupling rod; for the curved surfaces at E are merely parts of huge link pins having C and D for centres respectively. Therefore, for any one instant the angular velocities of wheels and cranks are identical.

Now, angular velocities of wheels A B are inversely as pitch circle radii, or as B G : A G. And angular velocities of links A and B are inversely as the divisions of the fixed link made by the coupler (see pp. 863–4) or as B F : A F. Hence B G = B F and A G = A F: the points G and F must coincide: and the normal C D must pass through the meet G.

P. 554. **Electric Transmission:** further notes.

Units.—Electric energy is now estimated by the product of E.M.F. (volts) and Current (ampères): the resulting units being *watts.* Thus:

$$\text{Energy in watts} = \text{E C} \qquad \text{and Electrical H.P.} = \frac{\text{E C}}{746}$$

$$\text{But H.P.} = \frac{\text{ft. lbs. per m.}}{33000} \qquad \therefore \underline{\text{Foot pounds per m.} = \text{E C} \times 45{\cdot}4}$$

It appears, then, that an electrical H.P. (E.H.P.) is 746 watts, or 33,000 foot pounds per minute, as before; and the only reason for the name 'electrical' is to indicate that it is power given off a dynamo or to a motor. A *kilowatt* consists of 1000 watts.

The Board of Trade unit for quantity of energy supplied is 1000 watts acting for 1 hour, or *1000 watt-hours*.

$$\therefore \left.\begin{array}{r}\text{Board of Trade}\\ \text{Electrical unit}\end{array}\right\} = \frac{1000}{746} = 1\tfrac{1}{3}\text{ H.P. for 1 hour.}$$

Efficiencies.—The losses in a dynamo are said to consist of two parts, the one due to *electric resistance* of the wire coils, and the other to friction, hysteresis, and eddies, called *stray energy*. Now, if we could imagine a dynamo whose only loss was that of current resistance, or $C^2R \div 746$ (see p. 553), E.H.P. being that *given out*,

$$\text{Electrical efficiency} = \frac{\text{E.H.P.}}{\text{E.H.P.} + (C^2R \div 746)}$$

Similarly we may suppose the only loss to be the stray energy, and the efficiency could then be reckoned on that alone. Take H.P. to be that *put in*, we might say

$$\text{Mechanical efficiency} = \frac{\text{H.P.} - \text{stray power}}{\text{H.P.}}$$

But, as in all dynamos, there are both electrical and mechanical losses, the nett efficiency, or

$$\text{Commercial efficiency} = \text{mech. effy.} \times \text{elec. effy.}$$

and thus we account for all losses. The H.P. supplied is often spoken of as mechanical H.P., while that given off is electrical H.P. Hysteresis is a heat loss caused by resistance to magnetisation and demagnetisation.

Example 67.—A dynamo is designed to produce 200 ampères of current at a pressure of 100 volts. State this in Board of trade units. Calculate also the mechanical H.P. required to drive the machine, and the E.H.P. given off, if the commercial efficiency is 85 °/ₒ.

$$200 \times 100 = 20{,}000 \text{ watts}$$
$$\text{or in one hour} = 20{,}000 \text{ watt hours}$$
$$= \frac{20{,}000}{1000} = \underline{20 \text{ Board of Trade units}}$$
$$\text{Again } 20 \times 1\tfrac{1}{3} = \underline{26{\cdot}5 \text{ E.H.P. given off}}$$
$$\text{and } \frac{26{\cdot}5}{{\cdot}85} = \underline{31{\cdot}2 \text{ M.H.P. to drive it}}$$

P. 568. **Live Rollers.**—Messrs. Crandall and Martin (Am. Soc. C.E.) experimented on rollers 1 inch to 4 ins. diameter, and 1½ ins. long. They found, with *four* crushing surfaces,

$$P = \frac{Q\,\delta}{\sqrt{r}}$$

$$\begin{aligned} \text{where } \delta &= {\cdot}0063 \text{ for C. I. rollers} \\ &= {\cdot}012 \text{ for W. I. rollers} \\ &= {\cdot}0073 \text{ for steel rollers} \end{aligned}$$

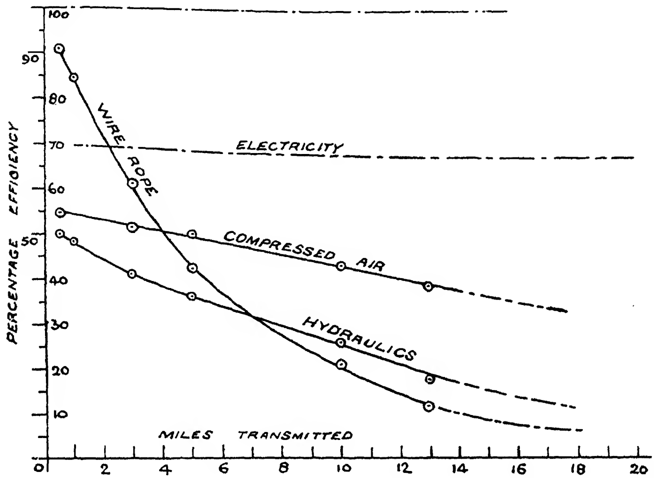
They also found the crushing load to be represented in lbs. by 880 × roller diameter.

580. **Efficiencies in long-distance transmission of power.**—The following table has been given by Beringer, shewing total efficiency of transformation and transmission, and, though probably open to some correction, is the best at present obtainable:

EFFICIENCIES PER CENT.

Distance transmitted in miles	Hydraulic Power	Compressed Air without re-heating	Wire Rope
½	50	55	91
1	49	54	85
3	41	51	61
5	37	50	43
10	26	43	21
13	18	39	11

By plotting the efficiencies to a base of distance transmitted, as in Fig. 885, we have a graphic means of determining the



Comparison of Efficiencies in Long-distance Transmission. Fig. 885.

relative advantages of the three systems. A probable line for the efficiency of electric transmission is also shewn. For very long distances it cannot be approached by any of the other methods.

A philosophical result of immense importance may be deduced from a consideration of this diagram. In wire-rope, *solid parts* are transmitted, and the efficiency falls off rapidly: hydraulics transmit *liquid parts*, and the efficiency holds up better: *gaseous parts* are carried in compressed air transmission, and the efficiency is considerably raised: but the greatest success of all is obtained by electricity, where a total efficiency of 67 % has been proved over a distance of 118 miles. These facts shew the electric current to be either etheric vibration or the passage of a very attenuated fluid.

CHAPTER X.

P. 600. **Value of Joule's equivalent.** — Rowland has repeated Joule's experiment of the churning of water, but on a much larger scale and for a longer time, driving the paddles by motive power and measuring the energy with a dynamometer. The much larger values thus obtained produce a greater approach to accuracy, and the altered value of J = 778 is now pretty universally accepted. Osborne Reynolds has since reviewed all known experiments, including those of Joule, and by careful correction deduces a result closely approximating to that of Rowland. As the correction is an increase of only ·515 of one per cent, the values within this book have not been altered. (*See p.* 1130.)

Now, on p. 603, C_v is deduced from C_p when J is known; conversely J may be found if γ is obtainable directly. This is not impossible, though difficult to do accurately. In books on physics we find

$$\left.\begin{array}{r}\text{velocity of}\\ \text{sound in air at } 32^\circ\end{array}\right\} = \sqrt{g\gamma H}$$

$$\text{where } g = 32{\cdot}2 \quad \text{and H} = \text{PV} = 26214 \quad \ldots\text{(p. 590)}$$

$$\text{Hence } \gamma = 1{\cdot}408 \quad \text{and } C_v = \frac{C_p}{\gamma} = \frac{\cdot 2375}{1{\cdot}408} = \cdot 167$$

$$\text{Now } K_p - K_v = c \ \ldots\ldots\ldots\ldots\ldots\text{(p. 604)}$$

$$\text{and } \frac{K_p}{K_v} = \gamma \ \ldots\ldots\ldots\ldots\ldots\text{(p. 603)}$$

$$\text{also } c = 53{\cdot}28 \ \ldots\ldots\ldots\ldots\ldots\text{(p. 590)}$$

$$\therefore \quad K_p = \frac{K_p}{K_p - K_v}c = \frac{K_p}{K_v\left(\frac{K_p}{K_v} - 1\right)}c = \frac{\gamma}{\gamma - 1}c$$

$$\text{But } K_p = JC_p$$

$$\therefore \quad J = \frac{\gamma}{\gamma - 1} \cdot \frac{c}{C} = \frac{1408}{\cdot 408} \cdot \frac{53{\cdot}28}{\cdot 2375} = \underline{774}$$

Prof. Thurston has pointed out a curious though totally unexplained coincidence, viz. that most probably

$$\gamma = 1 + \frac{4}{\pi^2} = 1{\cdot}405285$$

By using this quantity, $J = \underline{778}$

P. 625. **Nominal H.P. of Engines and Boilers.**—The proper method of stating power is undoubtedly by actual performance, and thus we obtain the I.H.P. and B.H.P. respectively. Nevertheless, although the older 'nominal' H.P. has fallen into bad odour through its gradually receding so far from the actual H.P., it is extremely convenient, for rating purposes, to have some rule for estimating the power of an engine, based on general dimensions, especially where an actual test is inconvenient. Neither can there be any objection to this, so long as the rule approximates to truth, and is never preferred to the latter.

Engine H.P. — The North East Coast Institution made a careful examination of marine engines and boilers in 1888, from which they devised a rule for *Normal Indicated H.P.* as they termed it, thus

$$\text{N.I.H.P.} = \frac{(D^2 \sqrt[3]{S} + 3H) \sqrt[3]{P}}{100}$$

where D^2 = added squares of every cylinder diameter, in inches.
S = piston speed in feet per m.
H = heating surface of boiler, in square feet.
P = working pressure in boiler, in lbs. per square inch.

Of course, this rule was based on the practice of 1888, and might require alteration from time to time, mainly on account of altered practice regarding expansion.

Boiler H.P.—By the horse power of a boiler we mean that power which would be developed if the boiler were to supply steam to an average engine. To make a simple rule, certain average constants must therefore be assumed, which are

C = coal burnt per sq. ft. of grate surface per hour, say 20 lbs.
E = water evaporation per lb. of coal burnt, say 8 lbs.
S = weight of steam used per I.H.P. per hour by the engine connected with the boiler, say 20 lbs.

Suppose, then, we require to know the I.H.P. obtainable from

three Lancashire boilers of 7 ft., 7′..6″, and 8 ft. diameter respectively, on the previous premises, we must first find G the grate surface. Taking all grate width at 6 ft., we have:

For 7 ft. boiler, grates about 6 ft. long, and area = 36 sq. ft.

$$\therefore \quad \text{I. H. P.} = \frac{G \times C \times E}{S} = G\frac{20 \times 8}{20} = 8\,G = \underline{288}$$

Similarly for the 7′..6″ boiler, grates about 6′..6″ long and area = 39 sq. ft.

$$\therefore \quad \text{I. H. P.} = 8\,G = 8 \times 39 = \underline{312}$$

And for the 8 ft. boiler, grates about 7 ft. long, and area = 42 sq. ft.

$$\therefore \quad \text{I. H. P.} = 8 \times 42 = \underline{336}$$

For other boilers the evaporation would depend upon the efficiency as compared with the above; and perhaps the steam used by the engine might be raised to 30 lbs. wt. per I. H. P.-hour.

P. 674. **Acceleration of Engine Piston**: Klein's construction.

When drawing an acceleration curve to a given velocity curve two cases present themselves, one where the base line represents time, and the other where it shews distance. The construction for the former, p. 860, is easily and correctly met by graphic differentiation. For a distance base, Proell's method, pp. 492 and 863, is very convenient for general cases, but when adopted for the harmonic motion of an engine piston, as at p. 674, it does not give absolute results at the dead points. We shall, therefore, here describe the construction due to Prof. Klein of America, which, though more cumbersome, and only applicable to the crank and connecting-rod, gives certain and accurate values at all points of the stroke.

Referring to Fig. 886, let AB be the crank and BC the connecting-rod. On BC as diameter describe a circle. Producing CB to D, describe a second circle having B as centre and BD as radius, and cutting the first circle in E and F. Then, EF will cross AC in G, and GA will be the *acceleration* of C, which

may be set up at C H. The same construction is followed when the crank is at B, and the piston at C_{11} viz. the circle on B C

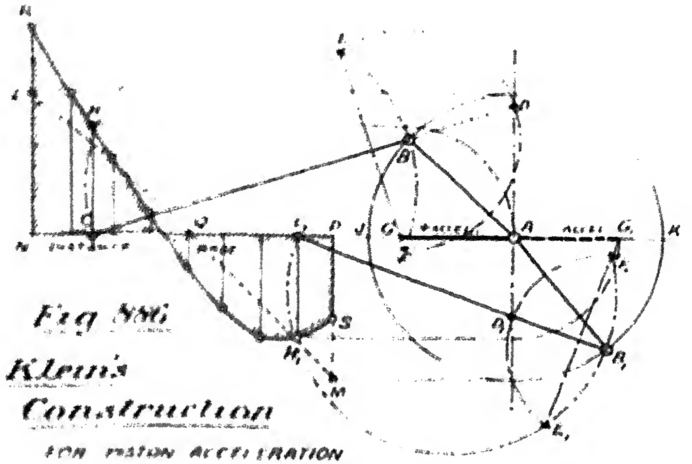


Fig 886

Klein's Construction

FOR PISTON ACCELERATION

will cut the circle of centre B_1 at E_1 and E_{11} and a line being drawn to meet A C produced gives $A G_1$ as the *retardation* to be set *down* at $C_1 H_1$.

The scale may be found from the following considerations:— whenever the crank pin reaches the dead points J or K, with tangential velocity v, the reciprocating masses are flung to left or right respectively with a force calculated from the formula $wv^2 \div gR$ (see p. 490). But inertia force is also represented by $wf \div g$ (pp. 473 and 674), where f = acceleration along J K

$$\frac{wv^2}{gR} = \frac{wf}{g} \qquad \text{and } f = \frac{v^2}{R}$$

for *pure harmonic motion* at dead points only. This condition is shewn by J N and K M, which must each equal $v^2 \div R$, thus shewing the scale to be used. Further, when a finite length of connecting rod is adopted, of length = c × crank arm, the end acceleration N E = $\left(1 + \frac{1}{c}\right)$ N J, and that at P S = $\left(1 - \frac{1}{c}\right)$ K M

P. 766. Calculation of Specific Volume of Dry

Saturated Steam, when the temperature, pressure, and latent heat are known.

In his 'Steam Engine,' Cotterill shows a very neat method of finding the specific volume V. Taking the $\theta\phi$ diagram, Fig. 887, for the Carnot cycle.

Calculation of Specific Volume of Steam

Fig. 887.

$$\text{Efficiency} = \frac{\tau_1 - \tau_2}{\tau_1}$$

Now heat expended per lb. weight of steam will be L_1, the latent heat at temperature τ_1.

$$\therefore \text{ work done per lb. weight of steam} = L_1 \frac{\tau_1 - \tau_2}{\tau_1}$$

which is also equal to the area of the diagram.

Now suppose $\tau_1 - \tau_2$ to be made very small, we have per lb. weight of steam

$$\text{Area of diagram} = \text{work done}$$

or difference of pressure × spec. vol. = latent heat × $\dfrac{\text{difference of temp.}}{\text{absolute temp.}}$

or
$$p(V - v) = \frac{L t}{\tau}$$

$$\therefore V = \frac{L t}{\tau p} + v$$

which is the same result as on p. 767; and the method of use is also as there described.

Pp. 611 and 767. **Engine Efficiency.** It is interesting to view the various losses in an engine and boiler system diagrammatically, so as to clearly demonstrate what are unavoidable and what may be improved. The case of a steamship has

been thus treated from end to end by Prof. Weighton, and by Mr. Leavenworth, and is here given in Fig. 888. Commencing at A with the heat in the coal as 100 % we find some 4 % of this lost

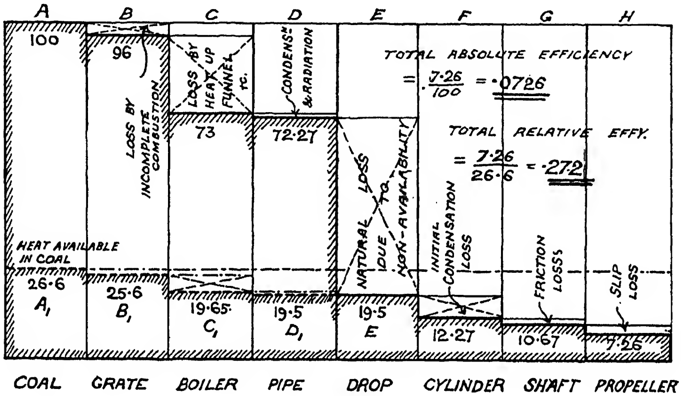


Diagram of Energy Losses

FROM BOILER FURNACE TO SHIP PROPELLER

Fig. 888.

at the grate B due to incomplete combustion or dilution. A further and very great loss occurs in the boiler at C on account of heat passing up the funnel, leaving us with only 73 % of the original heat to go to the steam pipe. A slight loss at D is caused by radiation and condensation in the steam pipe, and 72·27 % is given to the engine as dry steam. Now comes the great unavoidable natural loss on account of *unavailability,* present in all prime movers, whether driven by heat, wind, or water, so that the cylinder could not possibly give out more than ·27 % of the steam-pipe energy. On this amount, however, there is the loss by initial condensation, which is very large, so that only 12·27 % of the coal energy is 'indicated.' The final losses are the friction of the engine and propeller shaft, and the slip of the propeller, so that the ship is moved forward with only ·0726% of the original coal energy, supposing all the heat units to be available.

It does not appear, however, that the cylinder should be specially debited with the loss by unavailability, and more probably the best way is to deduct it from the coal energy in the first place.

The truth then is, that if the temperatures [illegible] boiler and condenser [illegible] the widest range [illegible] the

$$\text{Available energy} = [illegible] \text{ of total energy}$$

which means that the available energy in the coal [illegible] 30.6 [illegible] of the energy shewn by the calorific value. [illegible] the lower diagram, [illegible] shews the correct [illegible] the matter, and the cylinder is then [illegible] the condensation loss. In the upper diagram the greatest loss appears to be in the funnel, but in the lower diagram this is fought closely by initial condensation. The total efficiencies are ·0926 absolute, and ·272 relative, the latter being the most useful value for reference. This means that *practically* [illegible] *two thirds of the total available energy*. The figures are [illegible] as follows:—

Coal		Grate		[illegible]		[illegible]		[illegible]		[illegible]		[illegible]		[illegible]		
1	×	·96	×	·76	×	·90	×	·75	×	·63	×	·85	×	·68	=	·0926

·0926 ÷ ·306 = ·272 total relative efficiency.

Engine and Boiler Trials.—The object of experiments on engines and boilers is not only to find their efficiency, but to discover also the exact location of the various losses, in order to obtain data which may lead to their diminution. It is proposed to state here in general terms how such trials are conducted.

Whatever boiler or engine we are experimenting upon the procedure will arrange itself under (1) observations taken about every ten minutes, (2) calculations and conclusions, (3) diagrammatic statements, and (4) the balance-sheet of receipts and expenditure of energy.

Boiler Trials.—All the results are stated in terms of *one lb. of dry fuel*, and in thermal units. Also they may be reckoned from atmospheric temperature and pressure as datum lines, or 32° F. may be the zero employed. There are arguments in favour of each method.

BALANCE SHEET FOR BOILER.

Heat supplied	Heat used { both usefully and wastefully	Per cent.
To	By wasteful process:	
(1) Calorific value of one lb. of the fuel (wet) deducting water both internal & external)	(2) Loss by heating of water in fuel..........	
	(3) Loss in flue gases	
	(4) Loss by ashes and clinker	
	(5) Loss by unburnt fuel	
	(6) Loss by radiation and unaccounted for..........	
	(7) Evaporation of priming water	
	By useful process.	
	(8) Evaporation of water	
	....	100

(1) *Calorific value* must be found both by practical heat test and by chemical analysis (pp 697 and 906). For the latter, Mahler's empirical formula is often used to obtain greater accuracy than by the one on p. 906, thus

$$\left.\begin{array}{c}\text{Calorific value}\\ \text{per lb. wt}\end{array}\right\} = 14650\,C + 62100\,H - \left\{5400(O+N)\right\}$$

Where the letters indicate fractional weight per lb. of fuel. An intermediate value is finally taken, leaning to the side of the chemical analysis for preference

(2) *Loss by moisture in fuel* is that due to evaporation of water formed by union of oxygen and hydrogen when leaving the fuel (p 697, line 20), as well as to the free moisture. Such water absorbs 966 heat units per lb. wt., in addition to the heat required to superheat it to the temperature of the fuel.

(3) *Loss in heating flue gases.*—First, find weight of flue gases per lb. of fuel, either by direct measurement of draught speed with anemometer, or by comparison of flue gas composition with that of the fuel. By the former method, let

A = area of ashpit opening in square feet.
V = velocity of air through ashpit in feet per min.
t_2 = temperature of flue gases, found by pyrometer.
t_1 = temperature of stokehole.
τ_1 = absolute temp. of do.
P = pressure of atmosphere in lbs. per sq. ft.
w = weight of 1 cub. ft. of air at P in lb. units.
k = coal used per m. in lbs.

Then, AV cub. ft. of air per m. are used, whose weight per cub. ft. (*see* p. 590)

$$w = \frac{P \times 1}{53 \cdot 2\tau} = \frac{14 \cdot 7 \times 144}{53 \cdot 2\tau} = \frac{39 \cdot 4}{\tau_1}$$

$$\text{But } \frac{AVw}{k} = \text{weight of air per lb. of coal.}$$

And, as Total wt. × sp. ht. × temp. rise = total heat

$$\therefore \left.\begin{array}{c}\text{Heat units}\\ \text{wasted per lb. of coal}\end{array}\right\} = \left\{\frac{AVw}{k} + 1\right\} C_p(t_2 - t_1)$$

Note that C_p = specific heat of the flue gases at constant pressure and is almost always exactly ·25.

The second method may be briefly stated as follows :—

(*a*) Find % volumetric composition of flue gases, and convert at once into % weight composition.
(1 cub. ft. CO_2 at 32° weighs ·122 lb
1 cub. ft. N at 32° weighs ·078 lb.
1 cub. ft. O at 32° weighs ·089 lb.)

(*b*) Find weight of CO_2 per lb. of coal, thus :

$$\underset{\text{in coal}}{\%\,C}\,(1 \text{ lb. C} + 2 \cdot 66 \text{ lb. O}) = \underset{\text{in coal}}{\%\,C}\,(3 \cdot 66 \text{ lb, } CO_2) \ldots$$ (*see* line 23, page 687.)

(*c*) Hence find by proportion the total weight of each flue gas separately, per lb. of coal.

(*d*) From

$$C_p \times \text{wt. of each gas per lb coal} \times \text{rise of temp.}$$

the heat given to *each gas* may be found, and thus the *total heat given to the mixture* can be deduced per lb. of fuel, by addition.

(C_p = ·216 for CO_2, ·238 for air, ·244 for N, ·218 for O.)

(4). *Loss by ashes and clinker.*—This means the heat taken away when clearing the fires. It can only be reckoned by weighing the ash and clinker removed, and making a rough estimate of its temperature. If t_a = temp. of ash and t_1 that of stokehole, while ·25 is specific heat of ash, we have:

$$\left.\begin{array}{l}\text{Heat lost}\\ \text{in B.T.U.}\end{array}\right\} = \text{weight} \times (t_a - t_1) \times {\cdot}25$$

(5). *Loss by unburnt fuel.*—In analysing a lb. of fuel, the proper percentage of ash will be found. But the ash drawn from the grate is a greater proportionate amount, and the difference of the two will be the *unburnt carbon*, which may be stated per lb. of fuel by the heat units it would produce if burnt.

(6). *Loss by radiation and other causes* is obtained by subtraction in the balance sheet. It would be more correct to estimate the radiation and make a separate statement for 'unaccounted,' but this is never done.

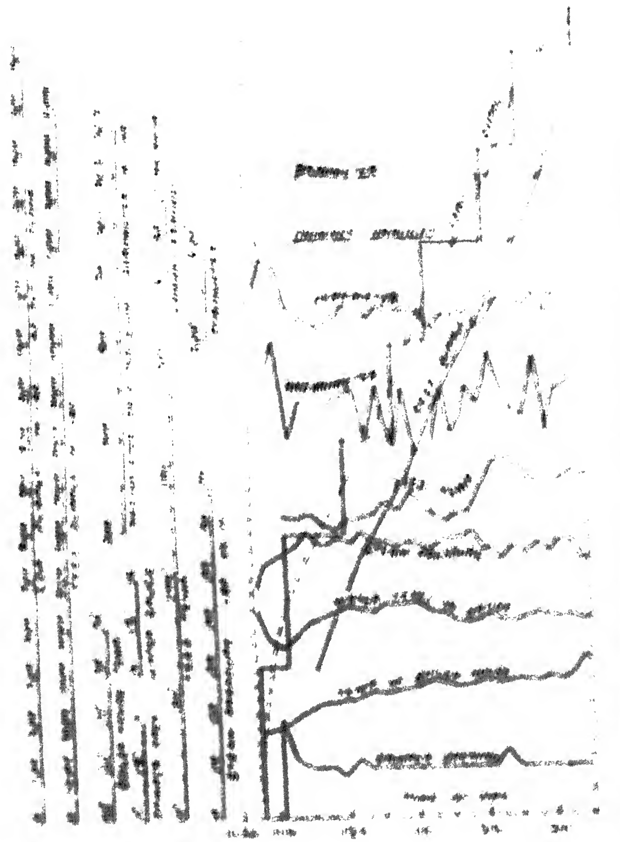
(7). *Heat used in the evaporation of water.*—This is the only useful application of the heat from the fuel. In the first place the quantity of water used, in lbs., must be obtained by measuring the feed supply. The latter is taken from measuring tanks, of which there should be two, one of which is filling while the other empties, and a careful noting of the time each tank is started is all that is required. Water-meters are sometimes used instead of tanks, but should be carefully tested. The total heat in the steam may be reckoned from stokehole temperature (p. 597). Then

$$\left.\begin{array}{l}\text{Heat used}\\ \text{per lb. of fuel}\end{array}\right\} = \frac{\begin{array}{c}\text{total heat}\\ \text{in 1 lb. of steam}\end{array} \times \begin{array}{c}\text{total}\\ \text{lbs. of dry steam}\end{array}}{\text{lbs. of coal used.}}$$

If the steam passing from the boiler be tested by the calorimeter (p. 878), a small percentage of the total weight will exist as entrained water (priming water), and this is not useful. Therefore the total heat in the dry steam and that in the priming water must be reckoned separately, and the latter will then be put down to wasteful process, and only the former to useful process. Lastly, it is usual, outside the balance sheet, to not only state the

actual evaporation per [illegible]
tion from and [illegible]

Equivalent evaporation [illegible]
from and [illegible]



Diagrammatic Statement of Data obtained in Boiler Trial (Fig. 889)

And the efficiency of the boiler may be given, being the evaporation heat ÷ heat in coal

Fig. 889 shews, diagrammatically, some actual data from a boiler trial. Steady gauge and steam pressure are of the highest

[illegible] and the [illegible] should be [illegible] half an hour before beginning and ending [illegible] kept bright at other times.

Steam Engine Trials. In these experiments we wish to measure the total heat given to the engine, and then to compare it with the indicated and brake energy respectively, the sources and amounts of the losses being determined on the way.

Balance sheet [illegible] Steam Engine [illegible]

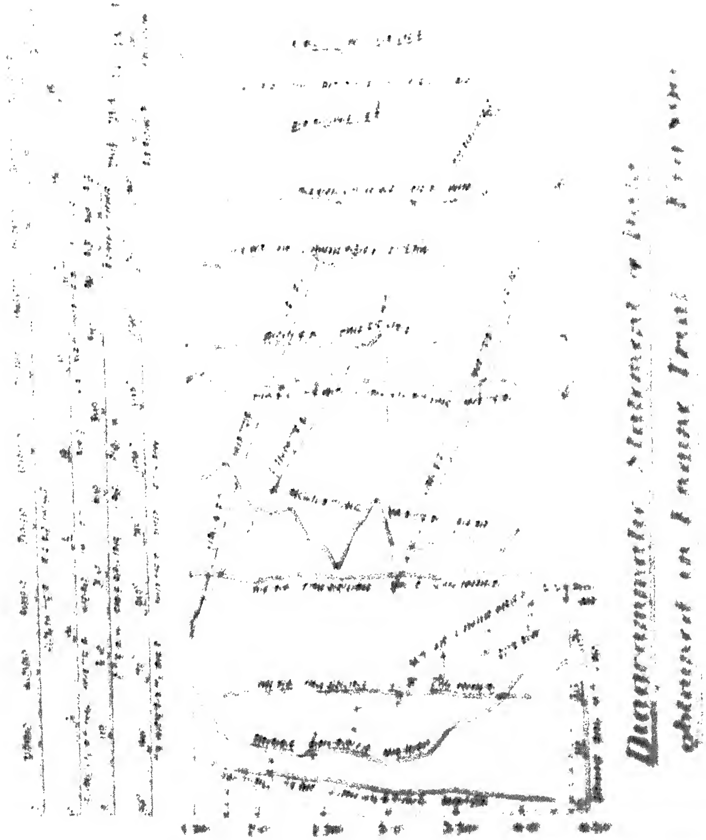
Heat supplied B.Th.U. per minute	Heat used usefully B.Th.U. per min. and wastefully	Per cent.
To: (1) Heat in dry steam as measured at engine stop valve	By wasteful process: (2) Given to cooling water (3) Left in condensed steam (4) Radiation and unaccounted (5) Left in jacket water	
	By useful process: (6) Converted into work in cylinder	
		100

Measurements are made every 10 minutes, and the calculations are from 32° F. as datum line.

(1) *Heat supply.* The weight of steam and water passing through the cylinder is found by condensing the steam and allowing all the water to be discharged into a calibrated tank. The rise in the tank will then shew the weight of wet steam used. But the total heat per lb. weight of steam will be

$$h_1 + xL_1$$

where [illegible]
[illegible]
[illegible]



thermometer fixed on the separator, just before the engine throttle valve; but very often the pressure gauge reading is taken, and the corresponding temperature found by reference to tables (pp. 598 and 777).

(2). ***Heat given to cooling water.***—The amount of cooling water used is most often measured by a water-meter, though tanks can be adopted also. Then the difference of temperature

of this water in passing through the condenser must be multiplied by the number of lbs. per m. to find the total heat used.

(3). *Heat left in condensed steam.*—If all the heat calculations were made from the temperature of the hot well, there would be no need to state this result, but, as we are measuring from 32° datum, the heat left will be

$$\text{lbs. of steam per m.} \times (\text{temp. condensed steam} - 32)$$

Note that, if the engine is non-condensing, a hot well or tank must be supplied, though not a condenser proper and air pump.

(4). *Radiation and unaccounted* is found by subtraction in balance sheet.

(5). *Heat left in jacket water.*—The jackets are drained regularly, and the weight of water obtained. Multiply this by the degrees of temperature above 32° to find the heat lost.

(6). *Heat converted into work in cylinder.*—This is the most important statement, and must be arrived at with great care. Indicator cards are taken every 10 minutes, and the I. H. P. calculated from the usual formula (p. 625). Then

$$\frac{\text{foot lbs. per m.}}{772} = \text{B.T.U. per m.}$$

The revolutions may be taken by counter or speedometer, or both.

All the time the experiments are proceeding we are measuring brake H.P. by absorption (p. 576 and 875) and are thus supplied with data for mechanical efficiency (p. 770). Finally, the results may be tabulated as follows:

(*a*) Thermal efficiency (p. 769, case *a* and reckoning from absolute zero)

(*b*) Efficiency of a perfect engine within same limits (*see* p. 769, where τ_1 = temp. live steam, and τ_2 = temp. hot well)

(*c*) Relative efficiency or comparison ratio (pp. 772 and 883)

(*d*) Mechanical efficiency.

The weight of dry steam per I.H.P. and per B.H.P. per hour should also be given, and indeed are usually the only results stated for commercial purposes. A graphic statement of the data, as in Fig. 890, conduces to clearness.

Gas-Engine Trials.—We shall commence with the

BALANCE SHEET FOR GAS ENGINE.

(Datum lines: 32° F. and atmospheric pressure)

Heat supplied in B.T.U. per m.		Heat used B.T.U. p. m. { usefully and wastefully		Per cent.
To: (1) Heat of combustion, due to wt. of gas and air passing per m. + heat already in gas and air...	......	By wasteful process: (2) Heat rejected in jacket water..................	......	......
		(3) Heat discharged in exhaust gases	......	......
		(4) Radiation and unaccounted................	......	......
		By useful process: (5) Heat converted into work in cylinder......	......	......
	......		......	100

(1). *Heat of combustion* is evidently

$$\text{Cub. ft. of gas per m.} \times \text{calorific value of 1 cub. ft. of the gas.}$$

The former is found by gas-meter, and the latter (about 650 B.T.U.) by analysis, or by means of a gas calorimeter. Add to this the heat already in the gas and air, above 32°, thus

$$\text{B.T.U. per m.} = \frac{\text{weight of gas}}{\text{and air per m.}} \times C_v \text{ of mixture} \times (t - 32)$$

The value of the C_v and weight of mixture is given in paragraph (3) following, and C_v is used instead of C_p because we are really comparing the heat given with that in an imaginary gas engine which receives at 32° and discharges at exhaust tempera-

ture. In such case, the volume of charge and products would, of course, be constant throughout.

(3) *Heat rejected in jacket water.* The quantity of jacket water passing is often found by a water meter. Then

H.R.J. per min. = $\frac{\text{lbs. water}}{\text{min.}}$ × difference of entering and leaving temp.

(4) *Heat discharged in exhaust gases.* We must first find the specific heat, C_v, of the products. If the usual proportion of air to gas be adopted, i.e. about [illegible] to 1 by volume,

C_v for entering air and gas together = ·1833
C_v for products leaving = ·1896

These are obtained from the empirical formulæ of Grashof.

For entering mixture

$$C_v = \frac{\cdot 169\,R + \cdot 326}{R + \cdot 875}$$

which assumes an average sample of coal gas and takes ·169 as the specific heat of air at 32° constant volume. R always stands for the ratio of air to gas by volume.

For leaving mixture

$$C_v = \frac{\cdot 1684\,R + \cdot 286}{R + \cdot 98}$$

The specific heat of air is taken at ·1684 for the higher temperature.

Sometimes C_p is required in order to arrive at γ. Then at exhaust

$$C_p = \frac{\cdot 237\,R + \cdot 443}{R + \cdot 98}$$

That gives ·258 for C_p of mixture, and γ = ·258 ÷ ·1896 = 1·36.

Next, we require the weight of mixture passing per minute. For this, two meters are used, one for the air and one for the gas as they enter the cylinder. Reduce these volumes to their equivalent at 32° from the formula

$$\frac{pv}{t} = \text{constant}$$

Now, the weight of [illegible] lb., and a cubic foot of [illegible] weighs [illegible]

Weight of gas [illegible] per m[illegible] [illegible] + [illegible]

Weight of air [illegible] per m[illegible] [illegible] + [illegible]

The total weight of [illegible] gives also the weight of products.

The exhaust temperature [illegible] so we have

B.T.U. per m[illegible] in exhaust gases [illegible] of products per min. [illegible] of products × (exhaust temp. [illegible])

(4) *Radiation and unaccounted loss* is reckoned in the same manner, by subtraction in the balance sheet.

(5) *Heat converted into work in cylinder.* The indicator diagrams are somewhat difficult to take, and a good indicator with stiff spring and short stroke must be used. The number of explosions should be found by a counting arrangement attached to the charging valve. Then

$$\text{I.H.P.} = \frac{pLaN}{33000}$$

where N = number of explosions, L the stroke in feet and p the mean effective pressure. Of course, the

$$\text{B.T.U. per m[illegible]} = \frac{pLaN}{778}$$

Further, the brake H.P. is obtained by a suitable dynamometer, and the following statements are made :—

Gas used per I.H.P. hour.
Gas used per B.H.P. hour.
Mechanical efficiency.
Actual thermal efficiency.
Efficiency of a Carnot cycle between same limits of temperature.
Relative efficiency or comparison ratio.

The theoretical thermal efficiency of any gas engine may be found from the formula

$$\eta = 1 - \gamma \frac{t_r - t_e}{t_i - t_a}$$

where the suffixes stand for release, exhaust, ignition, and atmosphere respectively.

The expansion curve is $pV^{1 \cdot 38} = C$, and the compression curve is $pV^{1 \cdot 3} = C$.

Oil-Engine Trials.—The balance sheet is of exactly the same character as that of the gas engine. Taking the items separately.

(1) *Heat of combustion* due to oil per min. is found as before, being about 19,000 B.T.U. per lb. wt.; and the heat already in the air and oil must be obtained as follows:

(*a*) $\begin{matrix}\text{B.T.U. per m.}\\ \text{in air}\end{matrix} = \cdot 169\,(t - 32) \times \text{lbs. per m.}$

(*b*) $\begin{matrix}\text{B.T.U. per m.}\\ \text{in oil}\end{matrix} = \text{specific ht.} \times (t - 32) \times \text{lbs. per m.}$

The specific heat of petroleum averages ·45.

(2), (4), and (5). These quantities need no further explanation.

(3) *Heat discharged in exhaust pipe.*—The exhaust gases consist of CO_2 and air. In addition, however, on account of the large proportion of hydrogen in the oil, there is a considerable quantity of water formed, amounting to nearly half the weight of the gaseous products. The method of procedure is somewhat similar to what obtains on p. 938. Thus:

(*a*) Analyse the oil to find % C and H.

(*b*) Find the % volumetric composition of the exhaust gases, and transform at once into % weight composition (*see a*, p. 938).

(*c*) Find weight of CO_2 per lb. of oil, from

$$\underset{\text{in oil}}{\%\,C}\,(1\text{ lb. C} + 2\cdot 66\text{ lbs. O}) = \underset{\text{in oil}}{\%\,C}\,(3\cdot 66\text{ lbs. }CO_2)\ldots\left(\begin{matrix}\text{p. 697,}\\ \text{line 23}\end{matrix}\right)$$

(*d*) Hence from (1 *b*) find total weight of CO_2 per m., and by proportion from (*b*) the total weights of the other gases per m.

(*e*) As at (*d*) p. 938 find the total heat *per m.* given to the mixture of exhaust gases, between 32° and the exhaust temperature.

(*f*) Find weight of H_2O per lb. of oil, from

$$\underset{\text{in oil}}{\%\ H}\ (1\text{ lb. H} + 8\text{ lbs. O}) = \underset{\text{in oil}}{\%\ H}\ (9\text{ lbs. } H_2O)\ldots\left(\begin{matrix}\text{p. 697,}\\ \text{line 24}\end{matrix}\right)$$

and from (1*b*) deduce the total weight of water per minute.

Finally, the heat given to this water will be the sensible heat from 32° to 212°, the latent heat at atmospheric pressure, and the superheat from 212° to pyrometer temperature $p°$, or

$$\begin{matrix}\text{lbs. of water}\\ \text{per m.}\end{matrix}\left\{ S_h + 966 + (p° - 32)\cdot 48 \right\}$$

The heat given to exhaust gases and water being added, constitute the quantity (3) on the right side of the balance sheet, and the results of the test may be stated as at bottom of p. 946

APPENDIX IV.

FIFTH EDITION.)

CHAPTER VII.

P. 324. **Pneumatic Hammer.**—This tool has been much improved since the description on p. 324 was first written, and is now used for general percussive work, such as caulking, chipping, and even riveting. The drawings in Figs. 891 and 892 are taken from Mr. E. C. Amos's paper before the Inst. Mech. Engineers in Feb. 1900, and illustrate a very satisfactory hammer. A is the working cylinder, B the piston hammer, D the tool, E the controlling valve, E_1 a steel seating for the same, F the handle, H the throttle-valve, I the throttle valve trigger, a_1 passage leading from e to the cylinder a, and always full of compressed air when throttle valve is open, a_2 passage from cylinder to top of valve chamber, a_3 passage from front end of cylinder to annular space e_3 in valve chamber, a_4 exhaust passage at rear end of cylinder leading to exhaust through valve interior, a_5 bye-pass from a_2 to cylinder, a_6 bye-pass from cylinder to a_7, a_7 exhaust passage from forward end of cylinder to atmosphere, e opening into valve bushing, e_1 opening into cylinder, e_3 annular groove in valve bushing, e_4 openings in valve E leading to exhaust e_6, e_5 central chamber of valve, e_6 exhaust to air.

Compressed air having been admitted at g_1 by pressing the trigger I, the fluid passes through e, and under the head of valve E, thus forcing the latter into the position shewn in Fig. 891. The air can then pass into the cylinder through e_1, and thus moves piston B forward into the position shewn in Fig. 892. The piston being reduced in diameter at b forms a chamber b_1, so that as the piston nears its forward limit of stroke, the air enters the chamber b_1 from the passage a_1, which is in direct communication with space e. At the same time the passage a_2 is opened to b_1, and thus the air passes back to the top of valve E, forcing it down into

READY TO STRIKE

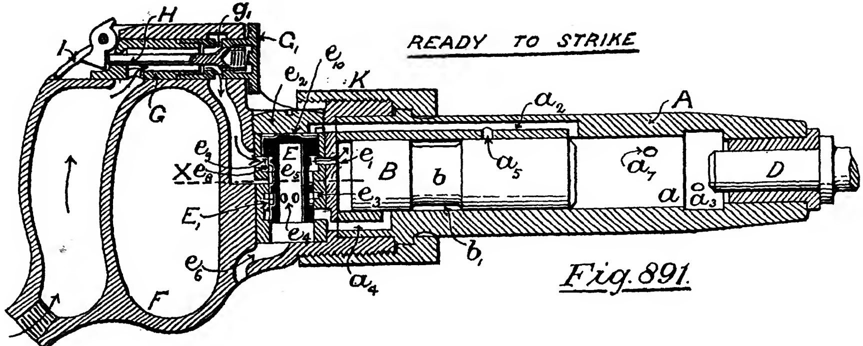
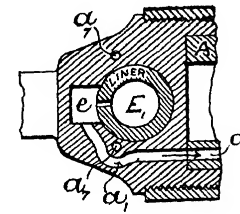
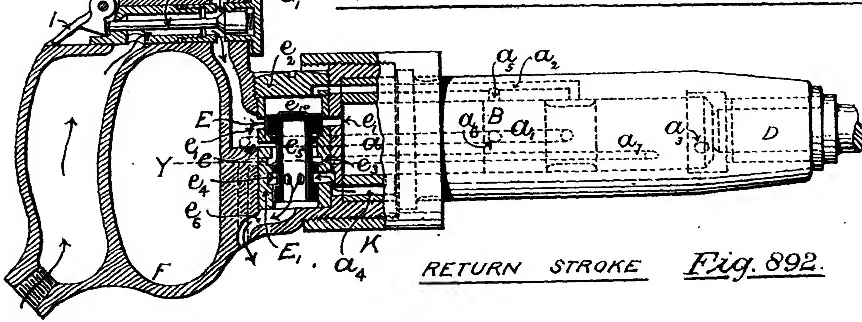


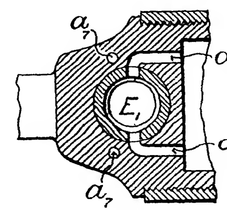
Fig. 891.



SECTION AT X.

"Little Giant" Pneumatic Hammer.





RETURN STROKE Fig. 892.

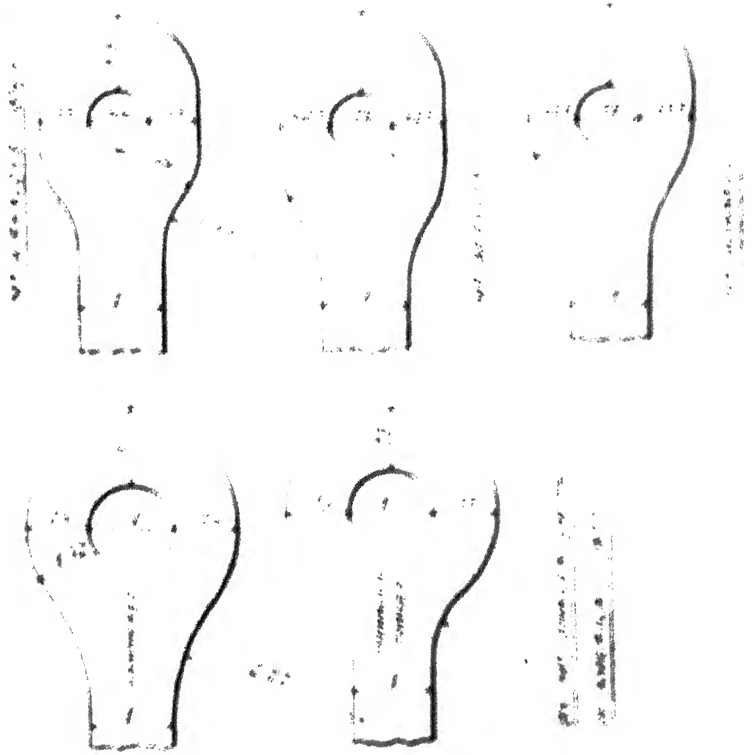
SECTION AT Y.

[illegible] the [illegible] are connected and [illegible] from to the [illegible] passage a_1 and through [illegible] When the [illegible] through [illegible] The valve [illegible] the [illegible] of the [illegible] the air is permitted to escape from the [illegible] through a_1 and [illegible] then by a_1 and [illegible] to the atmosphere. The pressure through [illegible] then again lifts the valve. The downward movement of [illegible] is due to the total pressure on the area [illegible] greater than that on the ring underneath the head. [illegible] is effected by cushioning, which in the [illegible] by the closing of port a_1 before the end of the back stroke. In the valve a little air is imprisoned round the [illegible] which [illegible] escapes through a_1 on the upward stroke. The portion c_2 of the valve is of a diameter nearly equal to the small bore of the valve bushing and there is also a small [illegible] groove at c_1. When the valve moves down, c_2 first enters the small bore of the valve chamber, and this tends to retard the passage of air through the bore, permitting the excess of air to act as a cushion.

Within certain limits the strength of the blow may be regulated by adjusting the amount of opening at the throttle valve [illegible]; this is produced by rotating the nut [illegible], and thus screwing the seating to left or right. Other hammers are made that are called "valveless," because they dispense with valve n, and use the piston to uncover ports in the cylinder walls, but such hammers are not suitable for heavy chipping or caulking. The speed of valve hammers is about 2,500 to 3,000 blows per minute, while "valveless" hammers vibrate at 10,000 to 20,000 strokes per minute, but with a lighter blow. The air pressure is about 60 lbs. to 80 lbs. per square inch, and it should also be mentioned that the "Little Giant" hammer is made by the International Pneumatic Tool Company.

CHAPTER XIII

§ 701. **Strength of a Suspension Link.** ... investigation on p. ... has been retained as ... duction to the theory of ... for which ...



Forms of Suspension Links as deduced from experiment

Fig. 803.

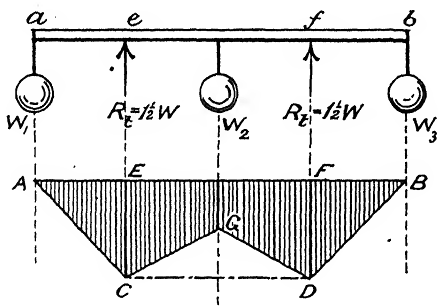
reliable, the student must be warned that it does not meet the proportions of suspension links adopted in practice. At the same time, as a satisfactory theory, taking into account an uneven

condition of stress at *a c*, Fig. 355, has not been propounded, we are obliged to accept such forms as are the result of direct experiment. In Fig. 893 are shewn six different forms of eye, all of which have been used on bridges, and all of which have been proved. In Mr. Shaler Smith's experiments the mode of manufacture was tested as well as the proportions, and therefore the results are probably the most reliable.

Pp. 443 and 850.

Example 68.—A beam is supported at two points one-quarter of its length from each end, and is loaded by three equal concentrated weights placed at the ends and the centre respectively; draw the diagram of bending moment.

The arrangement of the loads is shewn in Fig. 894, and the diagram of B_m is to be drawn on the subtractive principle. Firstly, construct the trapezium A B C D, where E C or F D = $W \times \frac{1}{4}l$. This



Beam Example. Fig. 894.

diagram shews the B_m which would be caused if W_1 and W_3 acted only, and the beam would be thereby curved upward. But W_2 will produce a reverse curvature on the beam between *e* and *f*, while not affecting the parts *a e* and *f b*. This moment, whose maximum is $\frac{W}{2} \times \frac{l}{4} = W \times \frac{1}{8}l$, or half that at E C, must be subtracted at every point between E and F, as shewn at C G D, and the remaining shaded figure is the true diagram of bending moment.

P. 471. **Wind Pressure on Roofs.**—Some years ago Prof. Kernot, of Melbourne University, constructed a blowing machine giving a fairly steady jet of air about one square foot in sectional area, and to this he exposed numerous models of

buildings, afterwards measuring the wind pressure on the various surfaces. He found that the only cases in which the ordinary methods of calculation for wind pressure on roofs (p. 471) agree with practice are :—

1. A roof on columns, with free air space beneath.
2. A roof lying on the ground, viz., without raised supports.

The other cases taken were :—

3. A roof supported on walls rising as far as the eaves.
4. A roof on walls having parapets above the eaves.

In case 3, no pressure was experienced on the roof whatever, while in case 4 there was actually a negative pressure, causing a decided tendency to lift. These experiments, the results of which are published in Vols. V. and VI. of the Australasian Association for the Advancement of Science, are supported by our general experience of roofs injured by wind.

CHAPTER IX.

Pp. 484 and 520. **Mechanical Advantage and Velocity Ratio.**—Taking the symbols on p. 481, the *theoretical form of the principle of work*, viz., neglecting friction, is

$$P \times d = W \times D$$

But if P_1 is the practical effort required, including frictional effect, the *practical form of the principle of work* becomes

$$P_1 \times d = W \times D$$

and therefore the *efficiency of the machine*

$$= \frac{P}{P_1} = \eta \text{ say.}$$

Now *Theoretical mechanical advantage* (T M A) $= \frac{W}{P}$; and *Practical mechanical advantage* (P M A) $= \frac{W}{P_1}$; and *velocity ratio* $= \frac{d}{D}$ which is numerically equal to T M A.

$$\therefore \text{ P M A} = \frac{W}{P_1} = \frac{W}{P} \times \frac{P}{P_1} = \frac{d}{D}\eta$$

$$= \text{velocity ratio} \times \eta$$

Suppose then a machine has a velocity ratio of 4 : 1, and an efficiency of 30 %, its real mechanical advantage would be

$$\frac{4 \times 30}{1 \times 100} = \underline{1 \cdot 2 : 1}$$

Referring to example 42, p. 484, let us assume an efficiency of 60 %. Then

$$\text{P M A} = \text{V R} \times \eta = \frac{\text{W}}{\text{P}_1} = \frac{1120}{60}$$

$$\therefore \text{V R} = \frac{1120 \times 100}{60 \times 60} = \frac{\text{follower}}{\text{driver}} \times \frac{15}{5}$$

$$\text{and} \quad \frac{\text{follower}}{\text{driver}} = \frac{1120 \times 100 \times 5}{60 \times 60 \times 15} = \frac{10 \cdot 37}{1}$$

So, in actual practice, the pitch line diameters may be 3″ and 31·2″ for pinion and wheel respectively.

Example 43 needs no correction, because this is a question of positive speed merely.

In example 44 let us again assume an efficiency of 60 %. Now T M A = 16 : 1 = V R

$$\therefore \text{P M A} = 16 \times \cdot 6 = 9 \cdot 6 : 1$$

$$\text{and } \text{W}_1 = 9 \cdot 6 \times 120 = \underline{1150 \text{ lbs.}}$$

If the Weston block has an efficiency of 35 % (see p. 577) we have

$$\text{P M A} = 20 \times \cdot 35 = 7 : 1$$

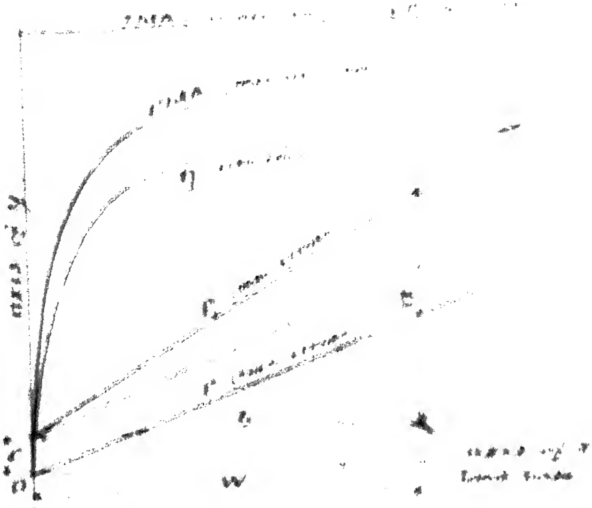
$$\therefore \text{W}_1 = 7 \times 50 = \underline{350 \text{ lbs.}}$$

In like manner the real pull on the handle in the screw jack, p. 520, with an efficiency of 60 % for the worm and 35 % for the screw, or a total of $\cdot 6 \times \cdot 35 = \cdot 21$ would be

$$\text{P}_1 = 20 \div \cdot 21 = \underline{95 \text{ lbs.}}$$

which means that 10 tons could not be raised conveniently.

Pp. 574 and 873. **Efficiency Curves.**—Whenever experiments are made upon machines in order to discover their frictional loss and efficiency under all loads, the full set of curves shewn in Fig. 895 ought to be plotted, to a load base The equations to



Efficiency Curves, Fig. 92

Line P is of the form $y = m\,x$, thus

$$P = \frac{1}{VR} \times W$$

Line P_1 is of the form $y_1 = m_1\,x_1 + c$, or

$$P_1 = \frac{a}{b}W + c$$

Next,

$$e = \frac{P}{P_1} = \frac{W}{VR\left(\frac{a}{b}W + c\right)}$$

But the construction of the machine will give us the velocity ratio, being

$$d \div D, \text{ so}$$

$$\text{TMA or VR} = \text{a constant}$$

And finally,

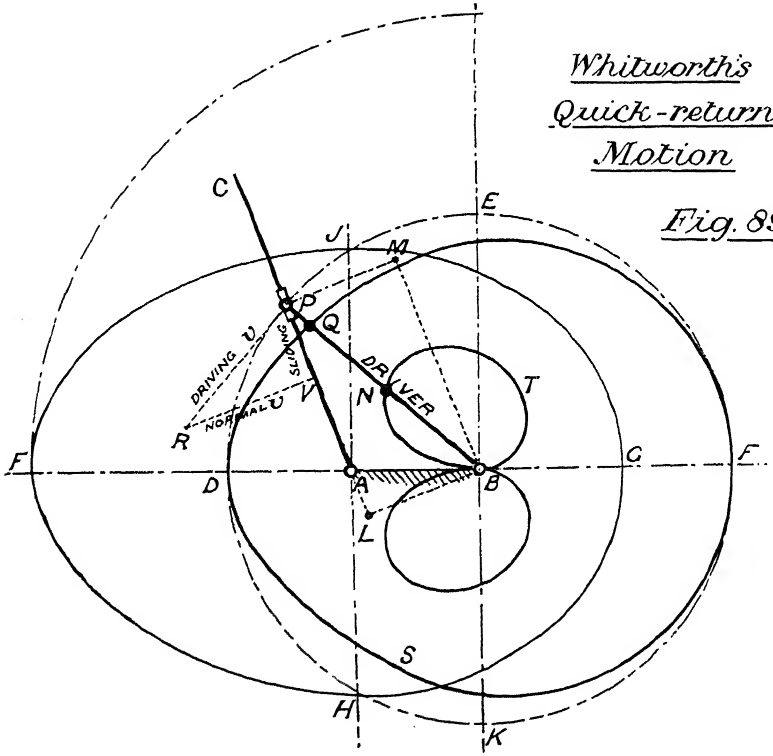
$$\text{PMA} = \text{VR} \times e = \frac{W}{\left(\frac{a}{b}W + c\right)}$$

In an engine, whether actuated by steam or gas, P would

represent the brake H.P., and P_1 the indicated H.P. These lines would be much more nearly parallel than in other machines, viz., the frictional loss would vary little from light up to full load. The efficiency curve would have the same general character, and the whole would be drawn to a base of load (B.H.P.).

P. 494. Whitworth's Quick-return Motion.

Example 69. Let two pieces move about fixed parallel axes in rubbing contact. Given the angular velocity of one of them, find that of the other: also the normal velocity, and the velocity of sliding at the point of contact. (Hons. Applied Mechanics Exam., 1897.)



Whitworth's Quick-return Motion

Fig. 896.

The first part of this question is answered by the construction shewn in Fig. 457, *p.* 493. The normal and sliding velocities of the slider are further shewn in Fig. 896. Referring to the latter figure,

A B is the fixed link, B P the driver travelling uniformly say, and A C is the driven link having a varying angular velocity. Following the method of *p.* 493, and considering B as the diagram centre, B P = velocity of driver at any point, and D E F K is a polar curve shewing ω of driver. Also G H F J is a polar curve shewing ω of A C by its radii vector.

Proceeding, let B P = linear velocity of P, and complete the parallelogram L P M B. Then with centre B, turn L round to N, and M to Q. Doing this for various positions, we have constructed the polar curve D Q F S, shewing linear *velocity of slider* P *normal to* A C. Similarly the polar curve B N T shews linear *velocity of slider* P *sliding along* A C. For proof, P R V is a triangle of velocities, drawn (1) right angles to P B, (2) right angles to A C, (3) parallel to A C, and it is easily seen that triangle B P L is similar.

P. 553. **Electric Accumulators or Secondary Batteries.**—The Planté and Faure cells have already been generally described. Both kinds of cells are still used, the E.P.S. being a Faure cell, and the Epstein a Planté cell, while the Chloride Co. claim to unite the advantages of both types. In the Faure cell the plates are lead grids, whose holes are filled with paste made as follows:—

Positive plate: red lead (Pb_3O_4) mixed with dilute sulphuric acid.

Negative plate: litharge (PbO) mixed with dilute sulphuric acid.

(Note: Comparing with a primary battery, the + is equivalent to zinc and the − to copper)

The probable condition in the Planté battery is:

	+ Plate	Electrolyte	− Plate
When charging ...	PbO_2	$+ 2\,H_2SO_4 +$	PbO.
When discharging ...	$PbSO_4 +$	$2\,H_2O +$	$PbSO_4$.

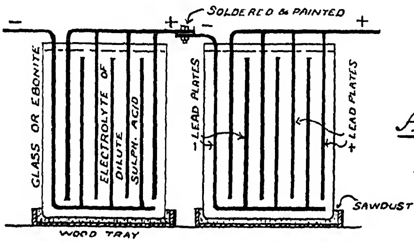
While in the Faure cell the reaction may be:

$$+ \text{ Plate, charging: } PbSO_4 + 2\,H_2O + SO_4 = PbO_4 + 2\,H_2SO_4$$
$$- \text{ Plate, charging: } PbSO_4 + H_2 = Pb + H_2SO_4$$
$$+ \text{ Plate, discharging: } PbO_2 + 2\,H_2SO_4 + H_2 = PbSO_4 + 2\,H_2O$$
$$- \text{ Plate, discharging: } Pb + SO = PbSO$$

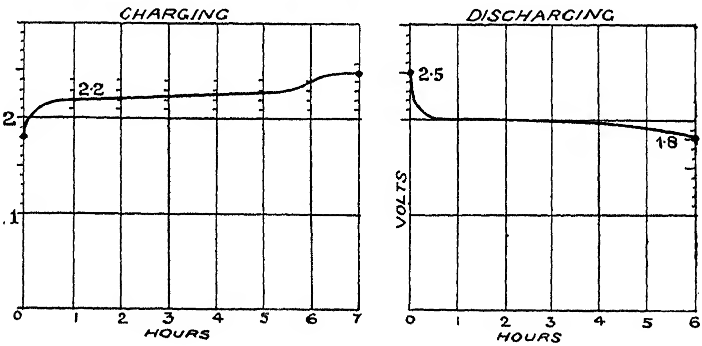
Fig. 897 shews diagrammatically the arrangement of the plates in a two-cell battery. The electrolyte or conducting liquid

between the plates is made up of 4 parts of sulphuric acid, having a specific gravity of 1·84, to 21 parts of distilled water, both by measure, the acid being slowly added to the water and the mixture allowed to cool.

Now, there are three kinds of winding in continuous current dynamos: (1) series wound, where the full armature current is



Electric Accumulators OR SECONDARY BATTERIES. Fig. 897



carried round the magnets to excite them, and then to the mains, all in 'series;' (2) shunt wound, where a portion only of the armature current is used to excite the magnets, carried in a small bye-pass wire or shunt; (3) compound wound, where both methods are adopted simultaneously. The shunt-wound dynamo is alone suitable for charging accumulators, except the excitation be caused by a separate small dynamo. When all is ready, the dynamo is run up to speed till the volts are properly adjusted, and only then is the circuit closed with the cells. The charging current should not exceed ·026 ampères per square inch plate of surface, and there should be a constant pressure whose volts =

2·75 × number of cells. The time of charging must be 12 hours when the cells are first filled, and may even be 30 or 40 with some makes of cells. Plates are of various sizes and areas, so that the charging rate may vary from 4 up to 15 ampères per plate. As soon as the cells are fully charged, the circuit is to be broken first, and the engine slowed down afterwards. Diagrams shewing variation of voltage during charge and discharge are shewn in Fig. 897, from which it will be seen that in charging the voltage gradually rises from 1·9 to 2·4, while at the same time the specific gravity of the liquid increases from 1·17 to 1·25, which is a further test of charge completion. If the charge is carried far enough, small gas bubbles fill the cell as they rise to the surface, and this effect is called 'milking.' Cells refusing to milk must be removed and examined. In discharging, the E.M.F. falls rapidly at first to 2 volts, and then gradually to 1·9 volts, after which no more current must be taken, there being only one-quarter of the original energy now left.

The charging and discharging times should be approximately equal, for although rapid discharge does sometimes take place, the plates not only deteriorate thereby, but a lower efficiency is maintained, due to heat losses caused by resistance. Also the most economical charging rate is one-half of the maximum allowable. It is important that the battery should be kept as fully charged as convenient—up to milky at least once per week. The discharge must never be complete, nor must the battery remain idle for any length of time, especially when low in charge; neither must the normal rates of charge and discharge be exceeded. The liquid may be replenished with pure water as it evaporates.

The *capacity* of the battery is measured in

$$\text{Ampère hours} = \left\{\begin{matrix}\text{average discharge,}\\ \text{ampères}\end{matrix}\right\} \times \left\{\begin{matrix}\text{total hours of}\\ \text{discharge}\end{matrix}\right.$$

And also in

$$\text{Watt hours} = \left\{\begin{matrix}\text{average}\\ \text{volts}\end{matrix}\right\} \times \left\{\begin{matrix}\text{average}\\ \text{ampères}\end{matrix}\right\} \times \left\{\begin{matrix}\text{total hours of}\\ \text{discharge}\end{matrix}\right.$$

Which may also be given in

$$\text{H.P. hours} = \frac{\text{Watt hours}}{746}$$

Capacities vary considerably, from 30 up to 600 ampères per cell, according to the number of plates and their size.

$$\text{Current efficiency} = \frac{\text{discharge ampère hours}}{\text{charge ampère hours}} = 80\% \text{ to } 90\%$$

$$\text{Energy efficiency} = \frac{\text{discharge Watt hours}}{\text{charge Watt hours}} = 60\% \text{ to } 70\%$$

Both these depend on rates of discharge, the lower figure for a rapid, and the higher for a slow rate.

Over-discharging, or too strong an acid, may either of them cause *sulphating* of the plate, the second fault being remedied by the addition of a small quantity of carbonate of soda. Buckling of the plate is caused by variable action, excessive rates, or by sulphating; and disintegration results both from sulphating and buckling.

For electric lighting, the

$$\text{No. of cells} = \frac{\text{E M F of lamps}}{2} + 2 \text{ or } 3$$

For traction, accumulators rapidly deteriorate through vibration or excessive rates (*see next paragraph*).

P. 554. **Electric Traction.**—There are four methods now in vogue for carrying the current from main to car motors:

1. Overhead line, with small trolley.
2. Conduit, with slot rail.
3. Surface contact, with temporarily live studs.
4. Accumulators on the cars.

The overhead or trolley system is by far the cheapest, and, being very little objectionable, is now most used. The main is buried in the ground, and the live wires, one for the up and one for the down line, are carried on suitable posts and brackets, being connected to the main through the hollows in the posts. The current is taken to the motor from the overhead trolley through a 'fishing rod' (*p.* 554), and the return is by way of the rail, whose several parts are connected by 'bonds' at the fish-plates.

The conduit system is so very expensive that it has only been adopted where appearance has had to be considered. The

current is taken directly from the main in the conduit and returned by the side rails (see p. [illegible]).

In the surface contact system, studs are laid along the track centre at short intervals, and these are only made "live" when a contact piece on the under side of the car passes over them.

Accumulators are, as already stated, excessively heavy as yet, though there are signs of a possible reduction of weight. They cannot compare with the trolley system as regards first cost, while the expenses of repair and renewal are very great. The great advantage of the system is the separation of the cars, so that a defect in one car cannot interfere with the rest of the system. For small cars they produce a completely vibrationless mode of driving which is remarkably pleasant, but the cell load makes a larger proportion of the total than in full sized tramcars. Thus in the electric cabs, the cells weighed 14 cwt. and the cab 11 cwt., while the people would not exceed 5 cwt. In large cars, on the other hand, while half the load is due to the car, the other half is equally divided between the passengers and the battery.

The working expenses per mile are, with horse cars, 10.[illegible]*d.*, with steam cars 10.3*d.*, or about the same, and with electric cars on the trolley system 4.8*d.*

P. 580. **Frictional Loss in Engines.** Another way of measuring this loss is to find the revs. per m. of engine at moment of shutting off steam, and ascertain energy of rotation, $Wv^2 \div 2g$, of flywheel or any further pulleys = E, say. Then

E ÷ no. of turns before stopping dead

= energy absorbed by friction of the unloaded engine per turn.

Also $\dfrac{E}{\text{turns}} \times N$ = work absorbed per m. when running.

And $\dfrac{E\,N}{T \times 33{,}000}$ = H.P. lost in friction.

Then as before

I.H.P. at any load − H.P. lost = B.H.P.

This is only true on the assumption that in engines the lines P and P_1 of Fig. 895 (B.H.P. and I.H.P. respectively) are nearly parallel, which is a fairly correct assumption.

CHAPTER X.

P. 700. **Electric Ignition for Gas and Oil Engines.**—The advent of the motor car has brought electric firing into great prominence, so that tube ignition has been practically abolished for all types of engines. At the present time three systems of obtaining the electric spark are practically used: (1) the primary or the storage battery, whose current pressure is intensified by means of an induction coil, (2) a magneto-electric machine giving a current of low pressure, (3) a small dynamo also producing a low-pressure current.

The first of these systems may be illustrated by Fig. 898, which shews the wiring adopted in the De Dion motors. Taking the parts in order we have, A the battery, B the primary wires, in circuit with the inner or primary winding P, J the induction coil whose secondary winding S is coupled to the firing plug, the current passing through the wire W and returning by the frame E of the engine, and finally the make and break hammer M, also in circuit with the primary winding. The battery consists either of four Leclanché dry cells giving about 6 volts at 3 to 10 ampères, or of two to three storage cells of 2 volts each. The induction coil consists of two windings, primary and secondary, as mentioned, a soft iron core Q, and a condenser C made of layers of tinfoil and waxed paper. The plug G being in place, the switch S is closed and the engine shaft turned round, causing the cam H to rotate also, which permits the hammer D to make and break contact at the proper time, thus enabling the cell current to pass through the induction coil, with the result that a secondary current of great intensity but low ampèrage is created in the wire W, and the spark passes at K, igniting the mixture in the cylinder. The ebonite block L may be altered by a rod, as shewn dotted, so as to change the firing position as regards the engine stroke, and this constitutes a hand governor of great convenience, producing various degrees of speed and power. The firing plug is shewn at N to a larger scale, the wire and rod W W being insulated by porcelain, and in some cases mica. The De Dion engine rotating at 1500 revolutions per minute does not permit of the attachment of a trembling hammer to the induction coil, so the spark is produced

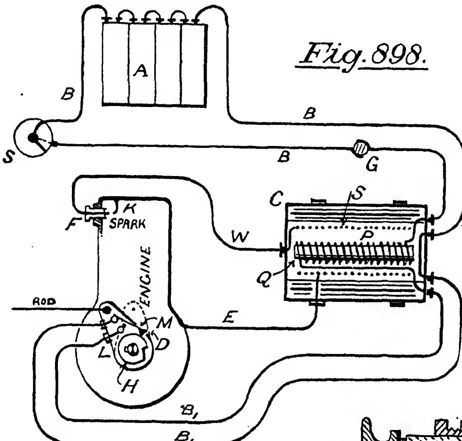
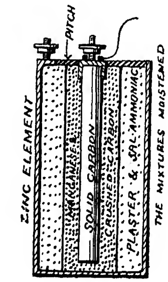
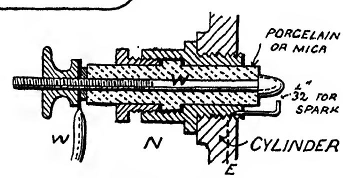


Fig. 898.



PRIMARY
DRY CELL
(LECLANCHE')



FIRING PLUG

Methods of Electric Ignition.

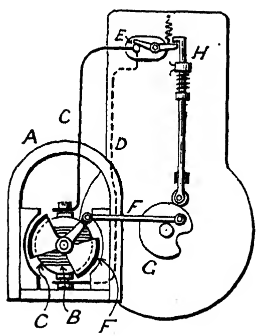


Fig. 899.

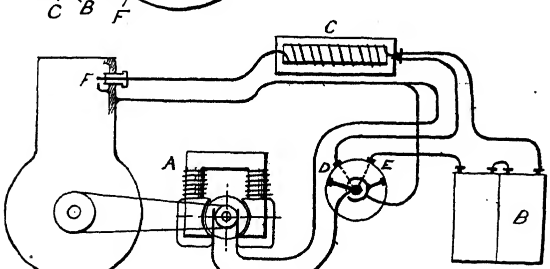


Fig. 900.

instantaneously in one flash. In other engines, like the original Benz, having but 800 revs. per m. or thereabouts, a trembling hammer on the coil causes a spark of longer duration.

The magneto-machine is represented by the Sims-Bosch gear shewn in Fig. 899. A is a permanent magnet of some strength, and B a fixed armature wound lengthwise with a wire C making circuit with the sparking plug E partly directly and partly through the engine frame D. F is a soft iron shield which is caused to rock on the armature centre by means of the connecting rod F coupled to a cam disc G on the engine shaft. As the cam rotates, it allows the tappet H to lower at the proper time and raise the trip lever J, thus breaking the armature circuit and causing the spark to pass at E within the cylinder. It is well known that the disturbance of a magnet's field will cause difference of potential in a circuit between its poles, and such disturbance is actually produced by the rocking of the soft iron shield.

Fig. 900 shews the method of connecting up a small shunt-wound dynamo A, which is driven from the engine shaft. As the current cannot flow till the engine is running, the latter must be started by means of the secondary battery B, which is provided with a primary coil C having large self-induction; but as soon as the dynamo becomes excited by the engine it is switched in circuit with the coil and firing plug, while the accumulators are fed in parallel by a portion of the dynamo current, so that the cells are kept constantly charged. Thus by closing switch E, we connect the battery B, coil C, and firing plug; and when ready the dynamo is coupled by switch D, sending some of its current by C and F, and the overflow through B. When the cells are fully charged, E may be opened.

CHAPTER XI

P. [illegible] Friction in Pipes.

Example [illegible]. Find (1) the head necessary to produce a flow of [illegible] ft. per sec. in a pipe [illegible] diameter and [illegible] long, the coefficient of friction being [illegible], and (2) the velocity of discharge of water through a pipe 1 mile long, [illegible] diameter, with [illegible] ft. of head, and the same coefficient.

Case I. Head lost in friction = [illegible]

[illegible]

Velocity head = $\frac{v^2}{2g}$ = [illegible] = [illegible] ft.

∴ Total head = [illegible] + [illegible] = [illegible] ft.

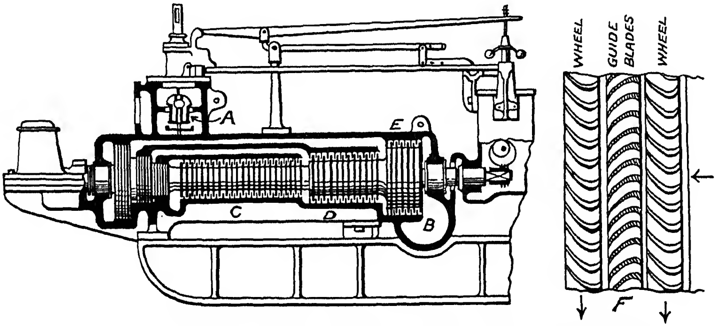
Case II. Total head = head lost in friction + velocity head

= [illegible]

[illegible] and v = [illegible] per second

P. [illegible]. Parsons' Steam Turbine. It appears that difficulties with regard to patent rights was the reason for the temporary abandonment of the axial-flow turbine, and the substitution therefor of the outward and inward flow turbine shown in Fig. [illegible]. Parallel-flow has since been returned to, and indeed the engines of the famous *Turbinia*, of [illegible] H.P., were of this type, both forms being now manufactured. Fig. [illegible] illustrates the parallel-flow type, and is easily understood. Steam enters by the valve, and traverses each of the chambers, [illegible], [illegible], and [illegible] in succession, the final

exhaust taking place at B; and an enlarged diagram at F shews the form of the blades in both wheel and fixed cylinder. (*See p.* 1168.)



Parsons' Parallel-flow Steam Turbine.
Fig. 901.

P. 898. **Balancing of Locomotives.**—Prof. Kernot, of Melbourne University, writes that he has got excellent results in the case of nearly one hundred locomotives, by first balancing the revolving parts perfectly, and next balancing $\frac{3}{4}$ of the reciprocating weights, *dividing the latter equally among all the coupled wheels*, so as to distribute the 'hammer blow' on the rail and make it harmless. The engines treated were mostly six-coupled inside cylinders of English type. (*See p.* 1201.)

APPENDIX V.

(SIXTH EDITION.)

CHAPTER I.

Pp. 34 and 785. **Moulding Machines.**—By the use of the hydraulic machine shewn in Fig. 902, which is made by Messrs. Bopp & Reuther of Mannheim, the pressing of the sand is performed simultaneously in both moulding boxes, and the removal of the latter effected at one operation, thus accelerating the work.

The hydraulic cylinder G contains a ram F, which supports a table U carrying the lower box E, and the constant pressure in cylinders S S balances the dead weight. The pattern plate A, to which are attached the two half-patterns B and C, is run into position along the rails shown. The upper box D is supported on a frame which can slide on vertical rods Q Q, and is connected to the ram F by the piston rod H, the crosshead J, and the ropes passing round the pulleys M N K L; so it follows that the raising of E will lower D, and *vice versâ.*

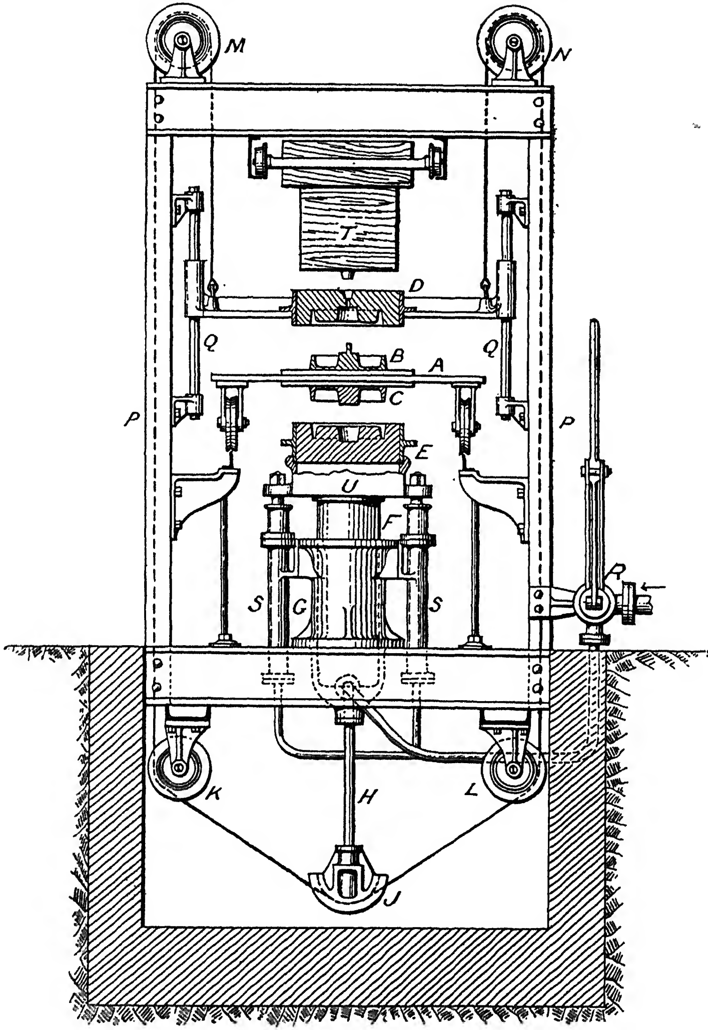
The pattern plate A being admitted, the boxes are filled with sand, and a suitable presser block placed at T. The pressure water is next let into the ram, and as E is raised to meet the plate, D is simultaneously lowered, after which E, A, and D rise together till they reach the block T; thus putting the required compression on the sand. The valve R being now opened to exhaust, the pressure is removed, and the ram again falls, lowering the pieces E, A, and D: together at first. The moment, however, that E detaches itself from plate A, so also does D begin to rise; and when they all take the positions shewn, the boxes may be removed and the operations repeated with a new pair of boxes.

CHAPTER III.

P. 86. **Soldering** is the process of uniting metals by readily fusible alloys that melt at temperatures that do not injure the work. The methods may be classed under *hard soldering* and *soft soldering*. The former, called also brazing, already described at p. 86, requires a temperature of 500° F. or more. Soft solder consists of 3 parts by weight of lead to 2 of tin, melting at about 340° F., and a flux of 'killed spirit' may be used to clean the joint and prevent oxidation. This flux is obtained by dissolving scraps of zinc in hydrochloric acid till the latter is completely changed to zinc sulphate, when an equal quantity of water is added. It is objectionable as causing rust, so in many cases, as in electric wiring, it is not permissible, resin only being allowed, to which a little oil may be added; but as resin does not clean the joint, the parts must be thoroughly rubbed and scraped before soldering. The soldering bit, made of copper, must also be cleaned and 'tinned' with solder before commencement. Soft soldering, when applied to unite bearing brasses and the like, is often termed 'sweating.'

CHAPTER IV.

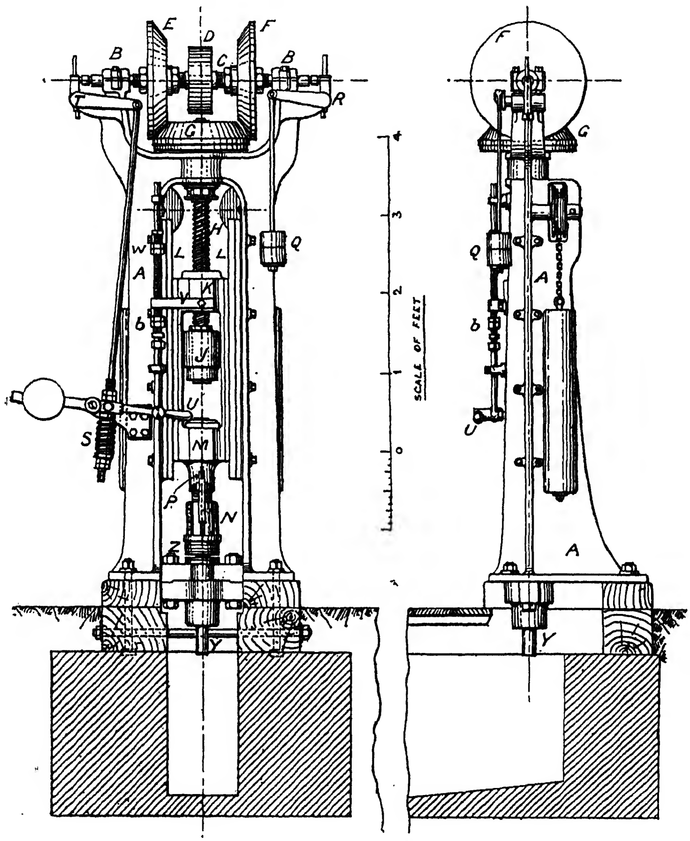
P. 95 & 282. **Rivet Making.**—Vincent's machine for heading bolts and rivets, shewn in Fig. 902*a*, is much in favour on account of the excellent work produced by it. A standard A carries in bearings B B a shaft C, which is rotated from a countershaft by the pulley D. Upon C are fixed two bevel friction wheels E and F, that alternately drive the third friction wheel G placed on the vertical screw shaft H. The coarse-pitched screw H carries an upper die J, which remains at a constant height, and also gives vertical motion to the tup K, which travels between the slides L L on the frame A. This tup is in one piece with the lower die M, into which is inserted the hot rivet shank; and the depth of the hole to receive the latter is decided by the position of the buffer nut N, which may be raised or lowered and then fixed by the screw clamp lever P. The shaft C is pressed leftward by the weight Q and bell-crank R bearing on the shaft end, and a



Bopp & Reuther's Hydraulic Moulding Machine.

Fig. 902.

rightward pressure is similarly caused by rod S through bell-crank T: thus wheels E or F are put in gear with G at will. The



Vincent's Bolt & Rivet Forging Machine.

(BY MESSRS. GREENWOOD & BATLEY) Fig. 902a.

starting handle U being raised, the weight Q puts F in gear, and H is rotated so as to lower the tup. The rivet shank is then inserted at M. The handle U is next pressed downward, putting

E in gear and causing the tup to rise till M and J meet to form the rivet head. The tappet V has thus reached the nut W and raised the striking bar X together with the handle U, compelling the tup to once more descend. But a stop rod Y passes upward through N and M to an arranged height suitable for the rivet length, so when the whole piece K M N strikes the buffer plates Z, the rivet is ejected by its meeting rod Y. The workman immediately inserts another shank piece, and the tup ascends automatically on account of tappet V meeting nut *b* and so depressing the handle J.

Pp. 124 and 748. **Case-hardening.**—An excellent form of animal charcoal is manufactured by Messrs. Palfreyman, of Liverpool, for case-hardening purposes. Pure white hard bone is cleared of pith and grease, and then carbonised in closed retorts at a uniform temperature. After removal, and when still hot, it is charged with a pure hydro-carbon oil, which prevents the accession of damp—a very great desideratum. The charcoal thus obtained is said to have three times the carbonising power of ordinary bone black: certainly much more than can be obtained with natural bone or leather, where the preparation has to be done in the boxes themselves; and the colour on the work is better and more uniform. The firm publish a pamphlet in which they make the following recommendations:—

Let the boxes be sufficiently large to allow 2″ space all round the work for pieces 4″ to 6″ diameter, for which boxes 12″ × 12″ × 8″ deep would probably be suitable; but for smaller work up to ¾″ screws say, the boxes may be 4″ × 4″ × 8″ deep. When laying the articles among the charcoal, press the latter lightly into the crevices or corners to ensure contact, and fill up the last 2″ with old or waste black that has been well dried: then cover with a suitable lid, dilute with clay, so as to resist the outlet of gas. The boxes being placed in the furnace, the heat is raised gradually, kept at cherry red for 3 hours for small articles, and at bright orange for 15 to 24 hours with large ones, a depth or 'case' of ⅛″ being produced after 18 hours of heat. For steel, 25°/. more black is needed than for iron.

If the articles are to be hardened, as is most usually required,

the boxes are lifted as soon as can be, and turned over so as to empty their contents into a tank of very cold salt water. To secure bright surfaces on the work, this dumping must be done close to the water surface, otherwise air is admitted, and dark blue or black is the result. It follows that pretty effects, such as streaked black or blue, may be obtained by suitable admission of air during cooling, which may be done both by dropping the pieces from a height of one to five feet above the tank, and by allowing air from a compressing pump to flow in with the water inlet to tank. The latter should be placed half-way up, and the outlet should be near the bottom. When colour is desired, the heat may be kept up for 3 or 4 hours with $\frac{3}{4}''$ work, but must not be overdone with any, and the colour effect will depend upon the amount of air admitted, or the height of drop. Finally, rinse the articles, wipe dry, finish in sawdust, and oil the surface well.

Sometimes carbonised though soft work is required, and the process is called annealing. Very old waste black is used, to which a little once-used black is added. The heat is only applied for a short time, and the cooling takes place very slowly, in ashes, charcoal, or waste bone.

To avoid the necessity of re-hardening, pieces of iron rod may be placed vertically so as to reach the centre of the box. These are withdrawn when believed to be heated for a sufficient time, cooled in water, and tested for depth of case or hardness. In spite of these indicators, however, very deep carbonising may need several heats.

The waste bone-dust is collected at the bottom of the cooling tank, after passing through a sieve half-way down the tank, and is thoroughly dried in ovens before re-using.

CHAPTER V

P. 142. **Cutting Speeds.**—Since this text-book was first written, a great change has occurred in the speeds allowable in machine tools. Practice varies considerably, some users adopting high speeds, with somewhat light cuts and fine feeds on the older tools, while others prefer more moderate speeds with deep cuts

and good feeds, such as may be obtained on newer and more strongly-built machines. Attempts are being made, and with some success, to remove all the surplus material at one cut, but to do so very powerful tools are required, and in all cases 'high-speed steel,' or at least good Mushet or self-hardening steel, must be adopted. As the whole question resolves itself into the cutting of a certain amount of material in a given time, it is advisable to compare one machine with another regarding its capacity to perform the work with rapidity; and to do this, the term 'power' of a machine tool has been introduced, thus:

$$\left\{\begin{matrix}\text{Power of}\\ \text{a lathe}\end{matrix}\right\} = \left\{\begin{matrix}\text{dia.}'' \text{ of largest}\\ \text{step of cone}\end{matrix}\right\} \times \left\{\begin{matrix}\text{width}'' \text{ of}\\ \text{step}\end{matrix}\right\} \times \left\{\begin{matrix}\text{velocity ratio}\\ \text{of back gear}\end{matrix}\right\}$$

The lathe in Plate V. would therefore have a

$$\text{Power} = 13 \times 3\tfrac{1}{4} \times \tfrac{42}{12} \times \tfrac{42}{12} = 518$$

and a very strong modern lathe would have a power of 750. To obtain high results the diameter of the largest step should be $2 \times$ centres, and the step should be wider than usual; the bearings should be parallel, and the front one have a diameter $= \frac{1}{2} \times$ centres, while its length should be $1\frac{1}{2} \times$ diameter; and the bed should have a total width $= 1\frac{3}{4} \times$ centres, being at the same time deeper than formerly.

The speed must bear a definite relation to the cut and feed. In a very powerful lathe, using high-speed cutting steel, the depth of cut being $\frac{1}{4}''$ to $\frac{1}{2}''$, and the feed $\frac{1}{16}''$ to $\frac{1}{8}''$, the speeds may be, for

Cast iron (with skin on)	60 feet per min.
Mild steel (35 to 40 tons strength) ...	70 ,, ,,
Wrought iron...	80 ,, ,,
Mild steel (25 to 30 tons strength) ...	90 ,, ,,

And with any good stiff lathe, using a good self-hardening tool steel,

Gun metal (ordnance mixture) ...	100 feet per min.
Gun metal (common)	120 ,, ,,
Brass	150 ,, ,,

But the diameter of the work is also said to affect the cutting speed. The figures given may be taken as true for a diameter

of 4 inches, with a ⅛ to 3/16 depth of [illegible] while greater diameters may have a heavier [illegible] A large and powerful lathe of [illegible] centres [illegible] take four cuts, each [illegible] depth and [illegible] feed [illegible] at 12 to 15 feet per min [illegible] 60 ft per m the diameter of the work being 14 [illegible] 150 ft per m may be obtained when [illegible] cut be no more than ⅛ and the feed [illegible]

Figs. 1497 and 1498. **Cutting Tools.** [illegible] largest amount of material at one [illegible] tool has been devised, called the "Gisholt" tool. It [illegible] made out of a bar of square sectioned tool steel [illegible] sharply to an angle of 90°, as shown at A Fig. [illegible] then grinding

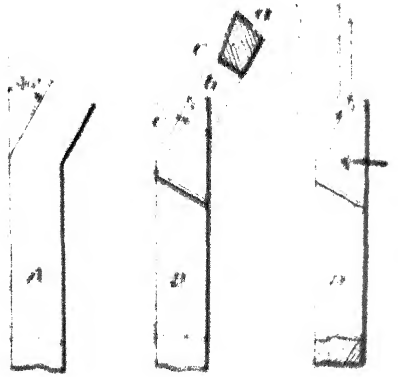


Fig. 1498.

Gisholt Tool.

off superfluous material till the form B is obtained. Top, front, and side rake are given at *a*, *b*, and *c* respectively, and the operation of cutting is lastly performed in the manner represented at D, where the saddle feeds in the direction of the arrow. Whatever the reason, there is no doubt of the rapid cutting property of this tool.

High-speed Cutting Steels. Mushet steel has already been mentioned at p. 799, as a type of self-hardening steel, that is, one that will harden automatically in the air after heating. This and similar steels are now almost exclusively adopted in

machine tools for cutting metals, because when heated by the frictional resistance of the cut, they approximately return to their original condition of hardness. But makers generally forbid the heating of the tool at any time beyond a bright cherry red, which is about 1500° to 1550° F. Now the higher we can allow the steel to be heated during forging, so much higher proportionally can it be allowed to overheat when cutting without drawing the temper, a fact which has led to the introduction by the Bethlehem Co. of cutting steel made on the Taylor-White process, where the higher permissible overheat gives increased value to the speed or the cut as may be desired. In this process the composition of the steel may be as follows :—

Chromium	Tungsten	Molybdenum	Carbon
½ to 3 %	1 to 6 %	1 to 6 %	·85 to 2 %

but the best results are obtainable when there is—

1 % Chromium with 4 % Tungsten ;

or, as an alternative,

1 % Chromium with 2 % Molybdenum ;

or, as a further choice,

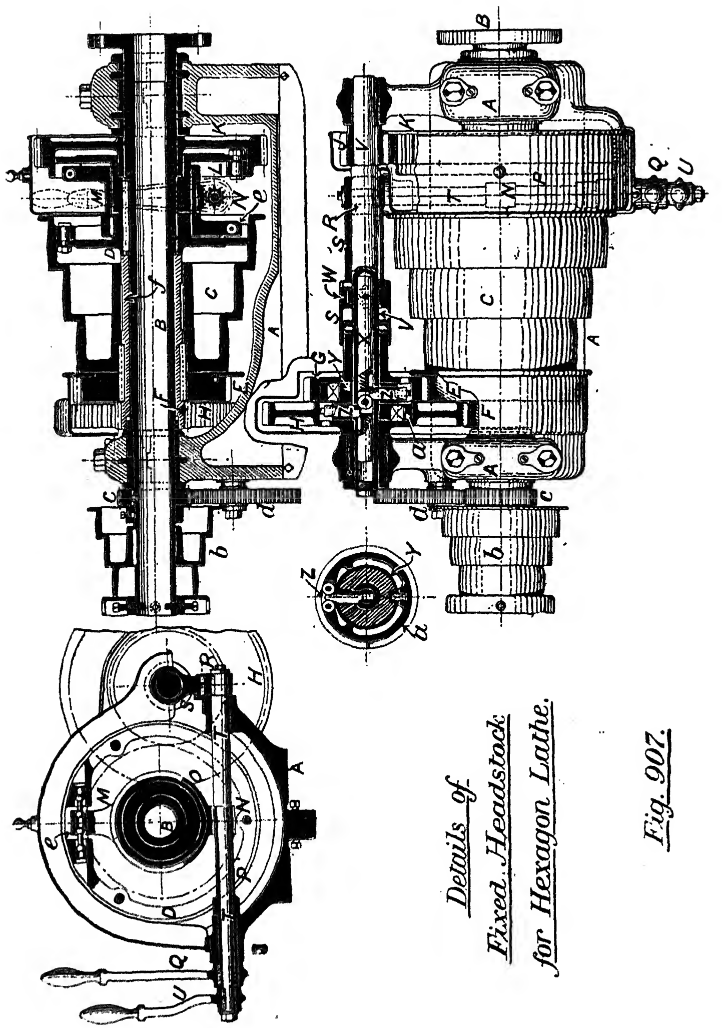
1 % Chromium, with
2 % Tungsten, and with
1 % Molybdenum,

the proportion of carbon within the limits given being immaterial. In addition to careful composition, the steel must receive a special treatment, which consists of raising the temperature (say during forging) to between 1500° and 1700° F., the surface being protected with powdered slag, then (after forging) letting down rapidly, though steadily, by placing in a bath of lead till a temperature of 1240° F. is reached. It is important that not the least rise of temperature be allowed at this stage, however brief. The heat should now be suffered to remain at between 700° and 1240° for the space of about five minutes, which may be done while cutting if desired; and with steel thus made and treated, a cutting speed of 150 feet per min. is easily practicable. The outer surface is generally damaged in heating, and should be ground off.

CHAPTER VI.

Pp. 200 and 814. **The Turret-head Lathe.**—A very excellent and powerful form of this useful tool is manufactured by Messrs. Alfred Herbert, Limited, that known as the 'No. 6 Hexagon Lathe' being illustrated in Plate XIX., Figs. 904–5–6, and in Fig. 907. Being strongly built it is suitable for any heavy chuck work that can be made in quantity, such as cylinder covers. Described generally, it is driven from a two-speed counter-shaft, has two sets of back gears, and two sets of change-wheel driving gear for the saddles. The saddles are two, one carrying the secondary or intermediate head with four tools for external screwing or bar work, and the other supporting the principal or hexagon head for internal work generally. The latter can carry six special bracketed tools of large size.

The Fast Headstock, shewn in Plate XIX. and in Fig. 907, carries in parallel bearings the hollow mandrel B, through which material of $2\frac{1}{2}''$ diameter can be fed. The cone pulley C, riding freely, has three steps of $13''$, $15\frac{1}{2}''$, and $18''$ diameter respectively, taking a belt of $4''$ wide, and has spur pinions E and H fixed to its left end, which gear with wheels G and H on the back shaft. The spur wheel K also rides freely on the mandrel, and gears with pinion J on back shaft. D and L are friction clutches fitting loosely in the cone pulley C and spur wheel K respectively, and keyed to the mandrel, while M is a clutch arm capable of sliding on a feather key. The arm M being moved lengthwise by the pinion N from the lever Q, acts upon toggles at R, and expands the clutch rings in D or L, so that the cone C *or* the spur wheel K may be fixed to the mandrel at will. In a similar manner the lever U may be moved to right or left, thus turning shaft T and sector R, and traversing sleeve S upon shaft V. A loose collar W and a pin connect sleeve S with the inner spindle X, and thus the sliding motion of S is transferred to the small roller which lies between the sliding wedges Z Z, causing one or the other of these wedges to be moved outward, expanding the clutch ring *a* in H *or* G. In this manner the spur wheels H *or* G are individually fixed to the shaft V, providing an immediate change from a back gear of 4·3 : 1 to one of 15·8 : 1. As there are two speeds on the



Details of Fixed Headstock for Hexagon Lathe.

Fig. 907.

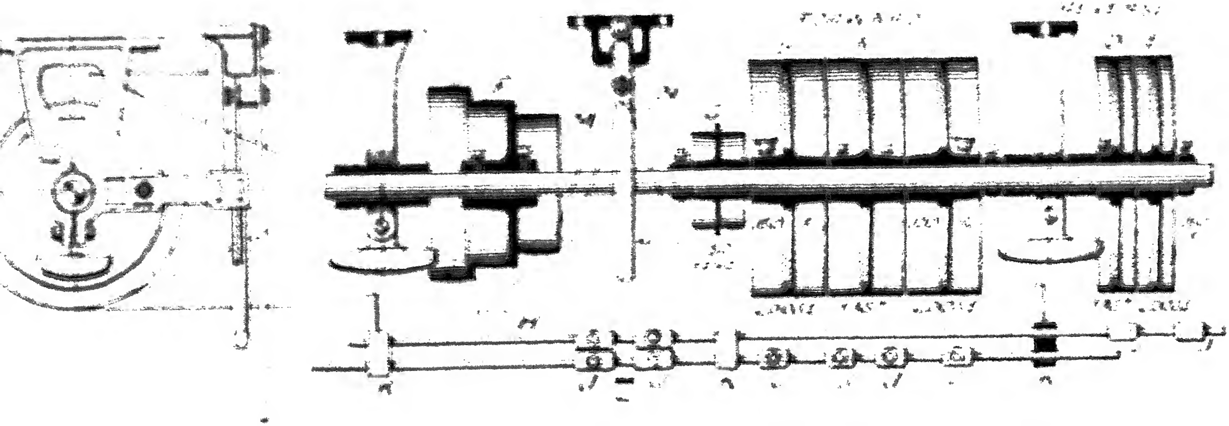
countershaft, it follows that there are six graduated speeds to each step of the cone, or 18 speeds altogether, changing from 4·5 up to 221 revolutions per minute without stopping the lathe.

The Countershaft [illegible] is given by two open belts, one of which is on the [illegible] pulley [illegible] and the other on the loose pulley [illegible] at 160 and the latter at [illegible] per [illegible] of these can be put in gear as required, by shifting the belt upon the fast pulley [illegible] The reversal is obtained from the crossed belt placed on [illegible], the loose pulley, the change being made to fast pulley at will, which then turns at the rev[illegible] The cone [illegible] transmits the power to the lathe, and [illegible] loose [illegible] for operating special tools, being driven at [illegible] for this purpose. The strap bars are shown in plan at [illegible], the forks at [illegible], and the bearings at [illegible], and the striking lever [illegible] operates them through stud [illegible] and second or fork lever [illegible]

The Change Wheel Gears are seen in Plate [illegible], one set being used for driving the leading screw *g*, and the other for the traverse shaft *h*. The former moves the mid saddle only, but the latter is connected to both saddles. The screw *g* is driven from the mandrel through the wheels [illegible] *d e f g h*, and *k*. At [illegible] there are three pinions of different diameters all coupled to wheel *e*, and the wheel *g* may be in gear with any of these. To effect the change the quadrant arm is lowered to put *g* out of gear, to slide it sideways, and to lift it into gear again, and thus ratios of [illegible] may be obtained. The screw shaft *g* is finally connected to the wheel *k* by a claw clutch *p*, which may be put in gear by handle *j* with bevel pinions [illegible] or [illegible], or may be between the two. When connected to [illegible] the driving is direct, but when coupled to [illegible] the pinion [illegible] is driven through an intermediate bevel wheel, and thus *g* is rotated forward or backward, or is stationary at will.

The traverse screw *h* is rotated from the mandrel through the cone pulleys *b* and *q*, and the pairs of wheels shown at *t u v*, and any pair of wheels may be put into gear by means of the handle *w*, which acts upon the internal spindle [illegible] and slides a cotter key [illegible] so as to fix *t*, *u*, or *v* to the shaft [illegible]. As before, a handle [illegible] puts a claw clutch in gear with bevel pinions [illegible] as to give forward or backward rotation or stoppage.

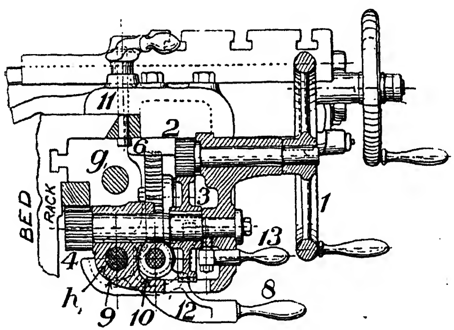
The Mid Head and Saddle.—This is spoken of as the chasing head, because largely employed in making external screws, but it



Countershaft Arrangement for Aexagon Lathe

Fig 908.

is also used on bar work generally. Its description will be understood by reference to Plate XIX. and to Fig. 908. It may be traversed in any of three ways: (1) by hand, (2) by self-acting shaft, (3) by leading screw. The hand traverse is obtained by handwheel 1, wheels 2 and 3, and rack pinion 4. The self-act is taken from shaft *h* through pinions 9 and 10, worm and wheel 5 and 6, and wheels 7, 3, and 4. The worm wheel is carried in a cradle 12, hinged round the shaft *h*, and held in position by a stop or trigger 13. Whenever the hand traverse or cross feed are to be used, the traverse worm 5 is released by dropping the cradle by handle 8, after releasing catch 13. The clamp 11 fixes the saddle for heavy tooling.



Hexagon Lathe.

Section through Mid Saddle.

Fig. 908.

As there are four tools to this head, so also are there four release stops to the traverse motion, each of which may be turned into position by hand. The stops are seen in Figs. 905–6, Plate XIX., being carried on a block projecting from the left end of saddle. They may be adjusted longitudinally, so as to strike the rod 15 at any given position of the saddle, the latter being also adjustable for the same purpose. The strike causes 16 to move a spindle to the right, and operate trigger 13, releasing the worm cradle and stopping the saddle.

A second worm 17 (Plate XIX.) drives a wheel 18, and thus the cross feed is obtained through wheels 19, 20, 21, and screw 22; or hand feed may be given by wheel 24, if trigger 23 be moved rightwards by hand to drop the cross-feed worm cradle. A bar 25 carries four stops 26, set in convenient positions, and each

of these can be presented upwards as required, by rotating the hand wheel 27. The vertical rod 28 has inclined ends so arranged that when the tappet 29 catches stop 26, and the stop bar is moved bodily, a grooved projection upon it depresses the rod 28, causing a bar 30 to move rightward, and so release the trigger 23. The worm cradle 17 at once falls, and the feed is stopped. A clamp 31 fixes the head for heavy machining.

Both worm cradles must be dropped when the leading screw is in gear. In Fig. 905, 32 is a slide carrying a half-nut 33, which can be moved in a direction across the bed. When lever 34 is in the position shewn, the nut 33 is in gear with the screw; but if 34 be lowered to a vertical position, it acts upon a coarse-threaded screw at 35, and causes slide 32 to move towards the operator, putting 33 out of gear with the leading screw. Several different guide screws are provided with each lathe.

All the motions mentioned are so interlocked that it is impossible for any two to be engaged simultaneously, a cam on bolt 35A preventing more than one cradle being raised at a time, and compelling both to drop before the screw can be put in gear.

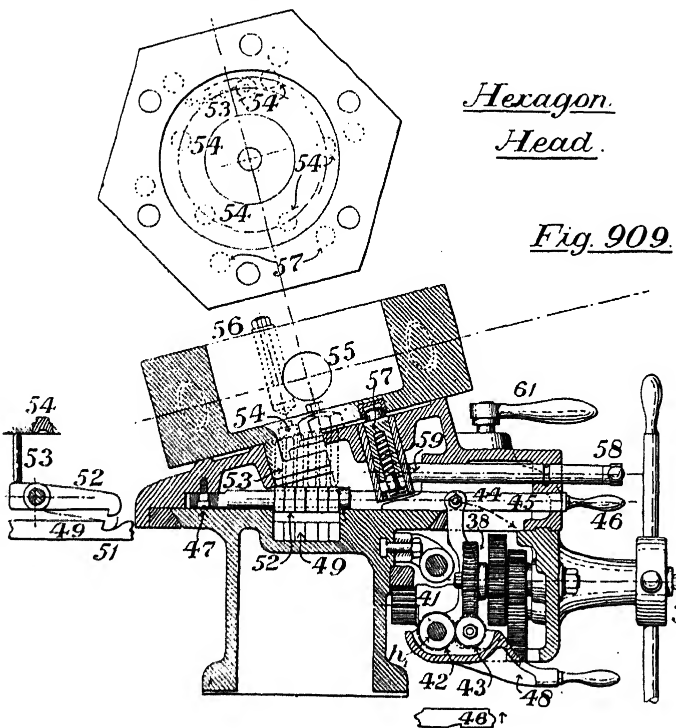
The Hexagon Head and Saddle is illustrated in Figs. 905 and 906, Plate XIX., and in Fig. 909. The saddle is moved by hand or power, the former by the 'pilot' wheel 36 through wheels 37, 40, 41 to the rack, and the latter from traverse shaft h_1 through spur wheels 42 and 43, worm wheel 38, and wheels 37, 40, 41 to the rack. When hand traverse is required, the self-act is put out of gear. At 44 (Fig. 909) is a notch on bar 45, shewn in plan at 46, and when handle 46 is pressed rightward, the bar turns on fulcrum 47, releasing stud 44–46, and causing the worm cradle to be dropped. The traverse may also be stopped automatically. At 49 are six bars fixed in suitable but independent positions by the screws 50, and having notches near their left end, as at 51, Fig. 909. There are six pawls 52, one to each bar, five of which are always held up clear of the bar by means of the props 53. One in turn, however, is allowed to rest on the bar, viz., that corresponding to the pawl in action, this permission being granted by the cup-shaped depression 54 on the hexagon head. As the depressions are specially spaced as shewn in plan of the head, Fig. 909, it is arranged that only the proper pawl shall drop into

its notch in the bar 49, when arriving near the end of the measured stroke, and the saddle still proceeding leftward, the bar 45 is relatively dragged rightward, again causing stud 45–46 to be released, and so to drop the worm out of gear. Bolted to the faces of the hexagon head are specially shaped tools, while boring bars may be passed through holes 55 and be held firmly by bolts 56. The head is rotated into its various positions by hand, the spring stud 57 deciding the same. The latter is withdrawn by a leftward turn of the handle 58, which operates a spur pinion 59 gearing with a rack on the side of stud 57. Fig. 905 shews two sets of gear at 42–43 which are changed as required by the handle 60. Finally, 61 and 62 are clamps for fixing the saddle when doing heavy work.

The *Chuck* is usually of concentric form: that is, the three jaws are operated simultaneously, but an independent chuck may be used if desired. Both of these have been described for other lathes at p. 154. The work is fed through the hollow mandrel B, and a constant stream of oil is poured upon it to take away the heat from the cut. The oil pump 63 is rotary and belt-driven, and 64 is a collecting trough for oil and shavings.

Special Tools and Holders.—As the Hexagon lathe is principally used for repetition work, it generally pays to spend some time designing the cutting tools to suit, and it is impossible to shew here all that can be done in such a direction. A few of the more important tool holders made by Messrs. Herbert are, however, illustrated in Fig. 910, all of which are intended for bolting to the hexagonal head. A is a 'dead' centre for converting the hexagon into a poppet head: it is provided with a fine adjusting screw. B shews a four-flute drill in its holder, and C a floating reamer, held by bolt *d* and set up by set screws *e* as may or may not be needed. D is a combination boring and facing tool. The bar is fixed in the holder by the clamp *j*, and is steadied in previously chucked and drilled work at *b*, the cutter being fixed at *c*. The facing cutters *a a* may be ground to any desired shape and fixed in heads on the holder, one or more of which, as *h*, are adjustable longitudinally. The turning tool holder E is most interesting. *h* is the cutting tool, which is ground to a broad

face, so as to act as a finishing as well as a roughing tool. For bar work the head must be traversed and more than one cut may be necessary, the tool being set deeper after each traverse until the required material has been removed. Narrow work is done without traverse by using a tool of the full width of work, and merely operating the depth feed. The latter is provided for by



Hexagon Head.

Fig. 909.

fastening the tool *h* in a box *j* which is capable of sliding in the bracket *k*, while the work is steadied by the rest *l*. The sliding feed is obtained by rotating the disc *w*, which advances a screw within the boss *n*, thus moving the rack *q* forward and rotating by wheel *r* the spindle *s*. At the bottom of *s* is a second spur pinion which gears into a rack *t* in the slide *j*. The depth of cut is decided by the inner screw *u*, which may be adjusted as

required and clamped by nut E. As the slide is moved forward, it is arrested when the screw W strikes the stop F. (See *p.* [illegible].)

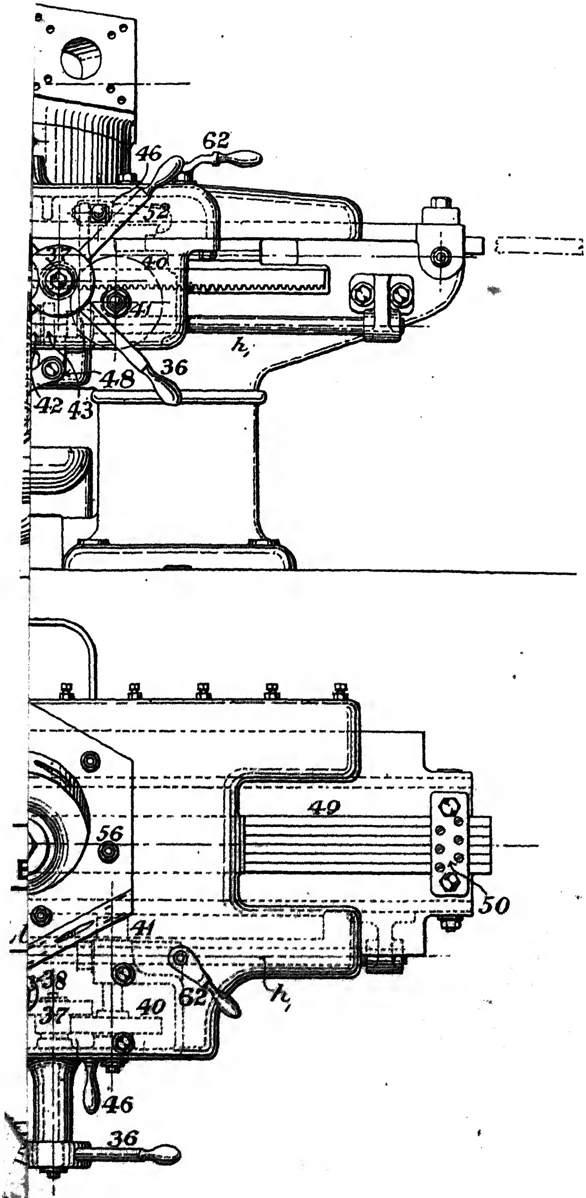
Pp. 256, 517 to 751, and 822. **Cutting Bevel-wheel Teeth.** It is now generally admitted that the only perfect way of cutting wheel teeth is to automatically 'roll' the blank at the moment of cutting, thus ensuring that the tooth shall be a true 'envelope' of the cutter's relative positions. This method has already been described for spur wheels at p. 822 and for worm wheels at p. 824; it remains to show a method of bevel-wheel cutting on similar principles.

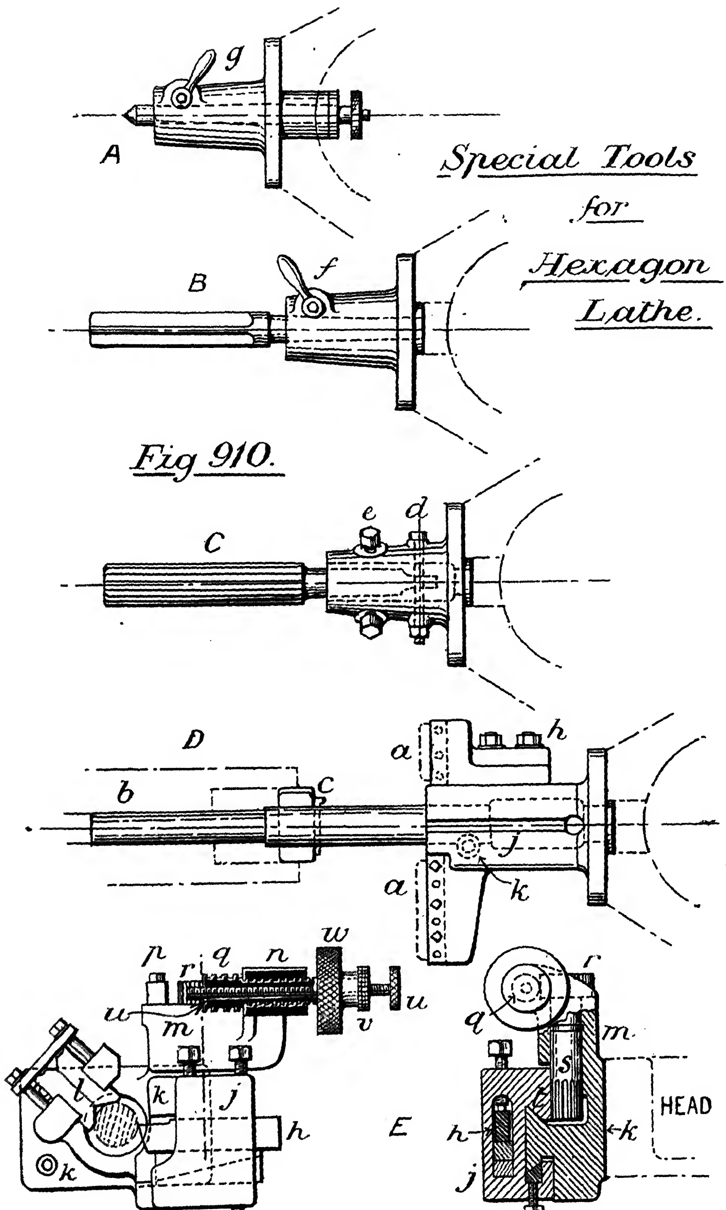
The 'Bilgram' bevel-gear planer is illustrated in Figs. 911-12-13, by the courtesy of Messrs. Pfeil & Co., who have introduced the machine from Germany, where it is manufactured by J. E. Reinecker & Co. Referring to Figs. 911 and 912, the wheel blank A is supported on an inclined spindle B, which is slewed round and rotated so as to cause a true rolling of A, while at the same time the cutting tool C reciprocates right and left so as to cut out the tooth interspaces.

The *tool mechanism* is that of a simple shaping machine, and is easily understood from p. 72, with the exception of the Whitworth quick-return motion, which may be seen at p. 133. The tool has to be lifted out of action after each cut has been finished, and this is provided for by the handle D, which rotates a lifting cam. E is a gauge bar, which carries a V piece and a pointer at G, for the purpose of setting the tool to correct depth, and to measure the amount of lateral feed. For different sizes of interspace, different thicknesses of gauge blocks are placed within the V. The lateral feed itself is given by the adjustable crank G, which reciprocates a rod H and slowly rolls the blank.

The *cutting tool* is seen in cross section at J, Fig. 913. It forms a portion of an imaginary rack, along the pitch line of which the blank is rolled, while the tool reciprocates in a direction at right angles to the plane of the paper, so that the tooth so cut in the wheel becomes an envelope of the rack tooth. It will be seen that the involute form of tooth has been adopted, because that is the only shape where one cutter is suitable for all teeth and for all cross sections of the tooth, the rack tooth being

PLATE XIX.





Special Tools for Hexagon Lathe.

Fig 910.

formed of straight lines at a constant angle. Refer here to Figs. 493 and 506, pp. 510 and 517. The true form being

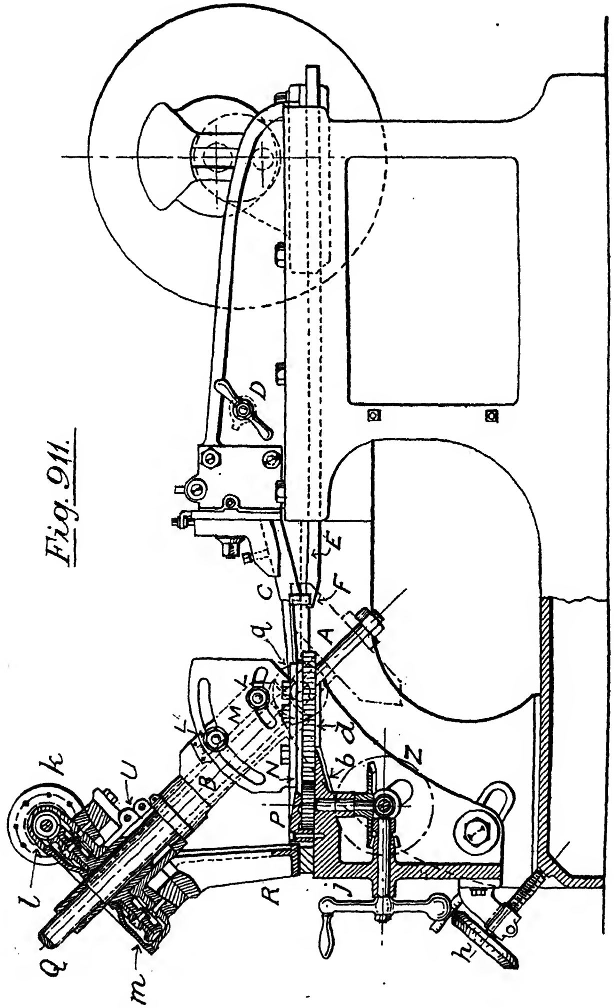


Fig. 911.

The "Bilgram" Bevel-gear Planer. ELEVATION.

obtained, the proper width of interspace is, of course, produced by lateral or rolling feed.

The Rolling of the Blank.—Referring to Figs. 911 and 912, the spindle B is set at such an angle that the root of the tooth in

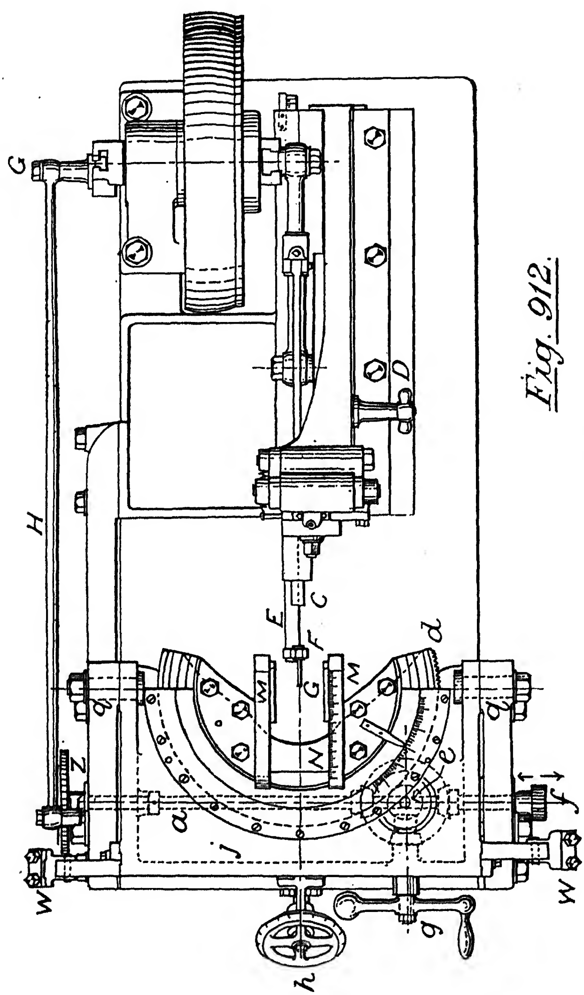


Fig. 912.

The "Bilgram" Bevel-gear Planer. PLAN.

the blank will lie in a horizontal plane with the cutter C, being so fixed by the bolts L L in the cheeks M M. The latter are carried

on a plate N which fits truly in circular V grooves P, and which may be rotated by the ratchet feed so that during rotation the upper end Q will remain at a constant height. But true rolling also requires the spindle to revolve while slewing, which motion is produced by the roll cone R (which is part of a true cone). Fig. 913 shews two views of the roll cone R, where S S is the base of cone of which R is a part, while T T the blank forms the opposite cone, and it follows that a true rolling of R will produce a true rolling of A. The revolution of B is secured by the coupling of K to B at U, and the true rolling of R without slip is obtained by connecting two steel bands V V to the brackets W W, which are fastened to points X X on the roll cone, the latter being supported at the correct height by the block Y. In practice, after choosing a suitable roll cone (of which twenty-two are supplied with each machine) for the particular angle of wheel, the cone and spindle are lowered at Y by one degree, such an adjustment giving a rather more rounded tooth and conducing to smoother action when the wheel is in use.

The *lateral feed*, as already mentioned, is obtained from rod H, which rotates the ratchet wheel Z, shaft *a*, pinion *b*, and the sector on plate N. Reversal is produced by the hand knob *f*, which couples either bevel pinion to shaft *a* by a clutch at *e*, and the handle *g* serves for hand feed or for setting. *Depth feed* is provided at *h*, where the turning of the screw will raise or lower the frame *j* round the fulcrum *q*, and thus simultaneously raise or lower the blank, the exact depth of tooth being read on the wheel *h*.

The *dividing* or pitching apparatus is seen at *k*, Figs. 911 and 913. It consists of the usual dividing plate *k* on a worm spindle *l* in gear with the worm wheel *m*. Between every complete cut the tool is lifted and the blank spindle rotated by the amount of the pitch.

The Working of the Machine.—After putting in the proper roll-cone, and setting to correct angle both for blank and cutting tool, the blank is lowered by the wheel *h*, and the tool begins to cut out the centre of the interspaces, while a gradual feed is being given at *h*, as shewn in Fig. 913, until the mid portion is cut out to the correct depth. During this operation the roll feed

is out of gear. The centre-cutting tool is next removed, a left-side cutter inserted, and the lateral feed coupled up for a relatively leftward motion of the tool: thus removing the full amount

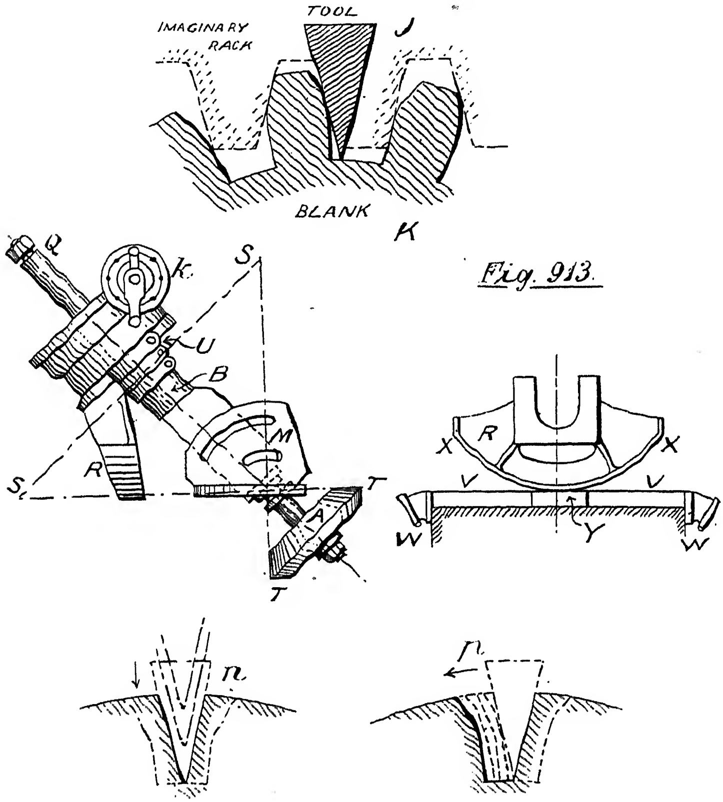


Fig. 913.

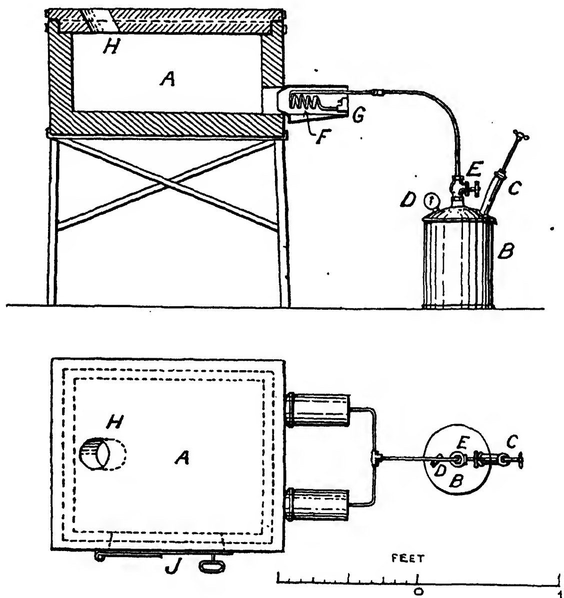
The "Bilgram" Bevel-gear Planer.

of the interspace on the left side, as shewn at *p*, Fig. 913; and this goes on, with the use of the dividing plate, until the left half of the interspaces of all the teeth are cut out. The spaces are again centred, and a right-side tool similarly used for the right

side of the interspaces, reversing the feed, when the cutting is complete. The tool used is a 'shaving' tool, which cuts only on its side edge, and which is only sharpened on its end, so that the shape may be preserved.

CHAPTER VII.

P. 284. **Rivet-heating Furnace.**—A convenient form of furnace is shewn in Fig. 914, as devised by Messrs. Fletcher,



Rivet-heating Furnace.

BY MESSRS. FLETCHER, RUSSELL, & Co.

Fig. 914.

Russell, & Co., where by the consumption of less than one gallon of petroleum oil per hour sufficient heat can be obtained to prepare from 300 to 400 rivets in the same time. The furnace

A is of fire-brick material, and the oil reservoir B is supplied with a compressing pump C, and a gauge D. When the cock E is opened, the fuel is fed forward under pressure till it passes through the hot coils at F, and finally emerges through the spraying nozzle G inducing in its passage to the furnace a sufficient quantity of air to complete the combustion. The flue is fixed at H, and the rivets are put in or taken out through the door J.

Pp. 339 and 828. **The Water-tube Boiler.**—A description of the 'Climax' boiler, Fig. 915, will prove interesting, not only because this boiler has shewn a high efficiency in actual practice, but because also of the totally new form of construction which it presents. It consists of a cylindrical vessel A standing vertically in the centre of the boiler, to which are attached a large number of horse-shoe-shaped tubes B C that assist circulation by reason of their lower ends C being much below their higher ends B. The fire space Q is annular in plan, and the fire is supported on a number of radiating bars. The boiler is surrounded by an outer casing U, lined with fire-brick, and there are four fire doors P P, four or more soot doors N N, and four ash doors T T. In erecting the boiler a brick foundation S is first laid, and after this a cast-iron plate R R, nextly the main cylinder A supporting the tubes, and lastly the casing U and the chimney. Access to the cylinder is obtained through full-sized manholes at E and D. The feed water enters at L, and may be either taken by V to the feed-water heater or economiser at H, and then by W and Z to enter the cylinder at X; or it may be fed directly through Y and Z to X, suitable cocks being provided to govern the direction of flow. At *a* is an automatic feed regulator, which is connected to a cock on the feed pipe (see p. 918). The pipe J for the outlet of steam, upon which is fixed the safety valve K, is taken from the top of the cylinder, where G is a dome space for dry steam, priming water being kept back by the deflector plate F. M is the pressure gauge.

The boiler illustrated is 11′ 2″ outside diameter, and 22′ 2″ in height of central cylinder, but may be carried much higher. The late Mr. Bryan Donkin made an exhaustive test of a 'Climax'

boiler, the dimensions of the boiler and the main results of the tests being as follows :—

Dimensions.

One central cylinder, 42″ diameter.

375 small tubes, 3″ diameter.

Outside diameter of casing, 11′ 2″.

Height of central cylinder, 22′ 2″.

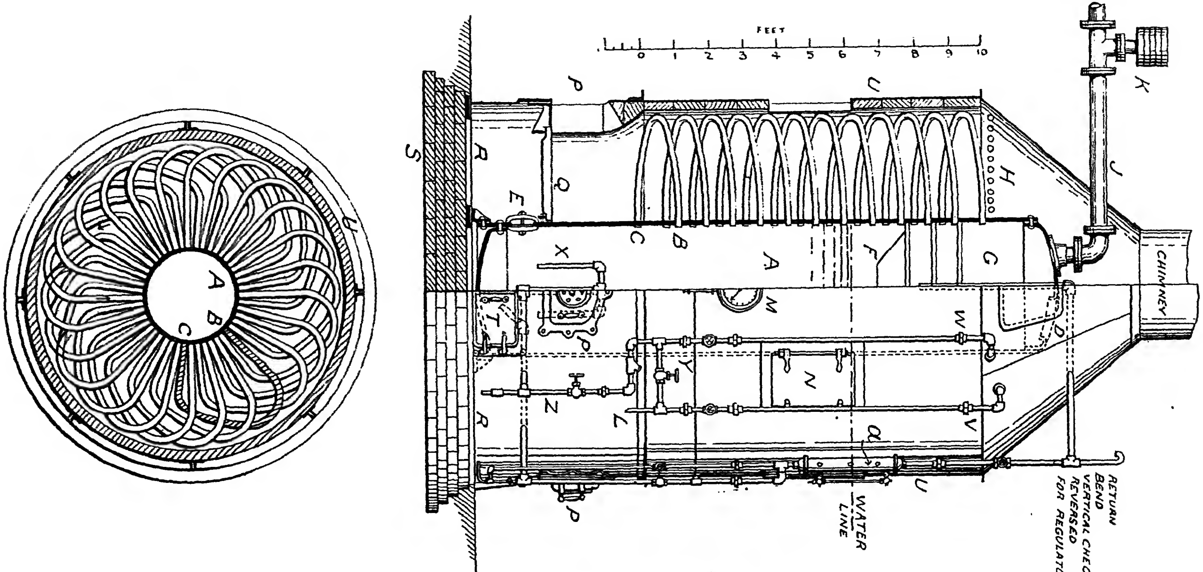
Total heating surface, 3129 square feet.

Feed heater consists of 180 ft. of $2\frac{1}{2}$″ pipe, area being 141 sq. feet, heating the water from $45\frac{1}{2}$° to 80° F.

The Test.

Duration	8 hours.
Dry coal per hour, best Welsh	1057 lbs.
Calorific value	14,886 B.T.U.
Temperature of chimney gases	593° F.
Feed per hour	10,234 lbs.
Steam pressure	192 lbs.
Steam superheated from	384° to 455°.
Horse power (at $34\frac{1}{2}$ lbs. evaporation) ...	297.
Thermal efficiency	79·31%.
Water evaporated per lb. of dry coal from and at 212°	12·22 lbs.

From these figures it will be seen that the boiler stands high for efficiency. At the same time steam can be raised very rapidly, a gauge pressure of 125 lbs. having been obtained in ten minutes, while in other cases sudden changes from 10 to 500 H.P. have been met without as much as a pound variation by gauge. Repairs are said to be slight, tubes being kept automatically free of deposit by reason of the rapid circulation, and all mud is easily removable from bottom of central cylinder, where it falls. Economy of floor space is, of course, self-evident.



The "Climax" Water-tube Boiler.

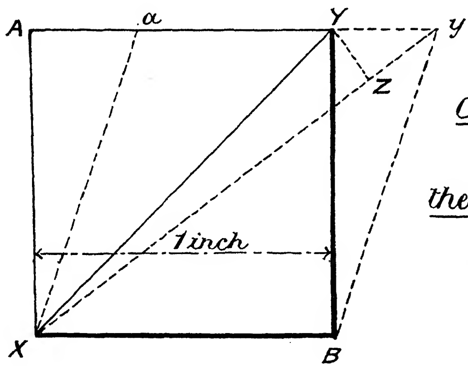
Fig. 915.

CHAPTER VIII.

P. 363. **Connection of the Moduli E, C, and K.**—Consider a cube A Y B X, Fig. 916, under the action of pure shear stress and strain, and let Y move to y *under shear.* Then X Y extends to Xy *under tension* (see pp. 366 and 755). But X Y, the original length $= \sqrt{2}$, and X Z $= \sqrt{2}$, and Yy is δ_s.

$$\therefore \text{ Total extension } = zy = \frac{\delta_s}{\sqrt{2}}$$

$$\text{and } \delta_t = \frac{\text{total extension}}{\text{original length}} = \frac{\delta_s}{\sqrt{2}\,.\,\sqrt{2}} = \tfrac{1}{2}\,\delta_s$$



Connection of the Moduli.

Fig. 916.

In like manner $\delta_c = \frac{1}{2}\,\delta_s$, and *a pure shear strain in one direction consists of two direct strains, compressive and tensile, each having a value half the shear strain,*

$$\text{or} \quad \delta_c = \delta_t = \tfrac{1}{2}\,\delta_s$$

But it has already been proved at pp. 366 and 755 that *two stresses, compressive and tensile, of equal intensity, cause a third stress in shear of the same intensity,*

$$\text{or} \quad f_c = f_t = f_s$$

Still considering the inch cube,

$$\text{Let M} = \frac{\alpha}{\beta}\ \text{(Poisson's ratio)} \qquad \text{and let } \frac{1}{\text{E}} = a$$

The three stresses being equal,

$$\text{Compressive strain} = f_c(\alpha+\beta)$$
$$\text{Tensile strain} = f_t(\alpha+\beta)$$
$$\therefore \text{Shear strain} = 2f_s(\alpha+\beta)$$
$$\text{and } C = \frac{\text{shear stress}}{\text{shear strain}} = \frac{1}{2(\alpha+\beta)}$$

Again, if the one inch cube be compressed equally on all sides,

$$\text{Compressive strain} = f\alpha - 2f\beta$$
$$\text{Volumetric decrease} = 3(f\alpha - 2f\beta) = 3f(\alpha - 2\beta)$$
$$\therefore K = \frac{\text{unit stress}}{\text{strain per inch cube}} = \frac{1}{3(\alpha-2\beta)}$$

Collecting the results, we have

$$E = \frac{1}{\alpha}$$
$$C = \frac{1}{2(\alpha+\beta)} = \frac{E\,M}{2(M+1)}$$
$$K = \frac{1}{3(\alpha-2\beta)} = \frac{E\,M}{3(M-2)}$$
$$M = \frac{\alpha}{\beta} = \frac{6K+2C}{3K-2C}$$

and the moduli are connected in value, if M be known.

Pp. 425 and 462. Strength and Stiffness of Shafts.

Example 71.—A steel marine engine weigh shaft has the dimensions shewn in Fig. 917, where A and B are the bearings, and the levers are as indicated. It is required to find the diameter of shaft in order that the stress shall not exceed 9000 lbs. per sq. in., the angle of torsion being limited to one degree in 20 ft. of length. (Actual case.)

$$\text{Load on reversing lever} = \frac{18\,(4000+4500+5000)}{15} = 16{,}200 \text{ lbs.}$$

and neglecting B_m, which is not considerable, greatest T_m is at centre, caused by the pull on I.P. and L.P. drag rods.

$$T_m = (4500+5000)\,18 = 171{,}000 \text{ lb. ins.}$$

The problem becomes :—given a shaft whose

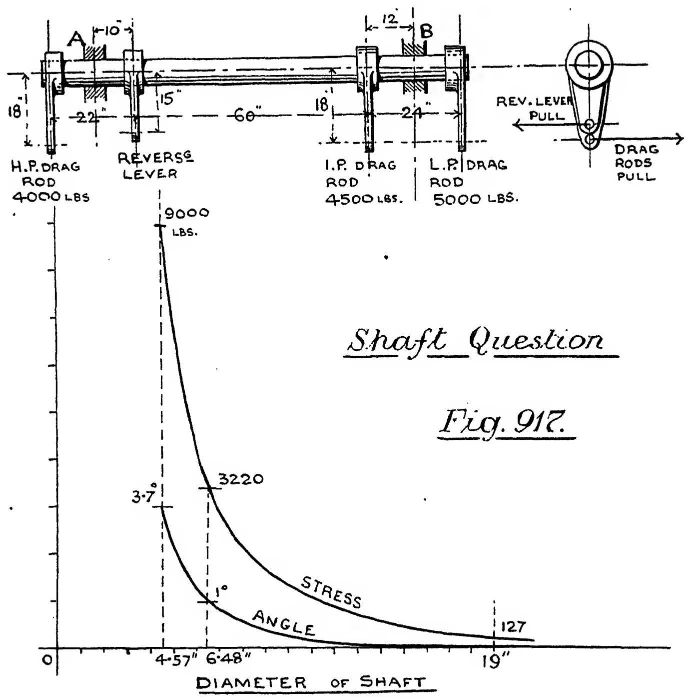
(*a*) Maximum $T_m = 171{,}000$ lb. ins.

To make this condition satisfy simultaneously

(*b*) Maximum stress = 9000 lbs. per sq. in.
(*c*) Maximum angle of torsion = 1 degree in 20 ft.

which it is impossible to satisfy fully. Taking (*a*) and (*b*)

$$d = \sqrt[3]{\frac{16\ T_m}{f\pi}} = \sqrt[3]{\frac{16 \times 171{,}000 \times 7}{9000 \times 22}} = 4{\cdot}57''$$



Shaft Question

Fig. 917.

Taking (*b*) and (*c*), as in Example 21, p. 425, with $l = 240''$,

$$d = \frac{2fl}{C\theta} = \frac{2 \times 9000 \times 240}{13{,}000{,}000} \times \frac{7 \times 360}{22 \times 2} = 19''$$

which means that if T_m be satisfied the shaft will be 4·57″ dia.; but such a shaft would be too springy, for

$$\theta = \frac{2fl}{Cd} = \frac{2 \times 9000 \times 240}{13{,}000{,}000 \times 5{\cdot}15} = {\cdot}0647$$

$$\text{or} = {\cdot}0647 \times 57{\cdot}4 = 3{\cdot}7 \text{ degrees.}$$

Hence 4·57 does not satisfy stiffness. We also learn that a shaft 19″ dia. is sufficiently stiff when stressed to 9000 lbs. per sq. in., but too stiff in fact, because our T_m cannot cause so high a stress, for

$$f = \frac{T_m \times 16}{\pi\, d^3} = \frac{171{,}000 \times 16 \times 7}{19 \times 19 \times 19 \times 22} = 127 \text{ lbs. per sq. in.}$$

We conclude that the proper course is to directly connect T_m with stiffness, for we shall not then exceed the required torsion angle; but the maximum stress will be much under 9000 lbs., which yet satisfies the conditions.

$$\theta = \frac{2fl}{Cd} \qquad \text{and } f = \frac{16\ T_m}{\pi\, d^3}$$

$$\therefore\ d^4 = \frac{2 \times 16 \times 171{,}000 \times 240 \times 57{\cdot}4 \times 7}{22 \times 13{,}000{,}000} = 1840$$

$$\text{and } d = \underline{6{\cdot}48''}$$

Also the stress produced by T_m will be

$$f = \frac{16\ T_m}{\pi\, d^3} = \frac{16 \times 171{,}000 \times 7}{22 \times 6{\cdot}48^3} = \underline{3220 \text{ lbs. per sq. in.}}$$

Answer:—*A shaft* 6·48″ *dia. will sustain a* T_m *of* 171,000 *lbs. without deflecting more than* 1° *in* 20 *ft. of length, the maximum stress not exceeding* 3220 *lbs. per sq. in.;* and the lower diagram in Fig. 917 will assist in shewing the peculiar conditions of the problem.

CHAPTER X.

Pp. 709 and 915. **Petrol-driven Motor Cars.**—There are now three practical sources of power for motor cars: steam, electricity, and oil or spirit. Steam has found favour on account of the highly regulable character of the working gas, the dispensing with spur changing gear and its noise and shock, and the reduced cost of repairs through freedom from vibration. On

the other hand, the cost of fuel, which usually must be oil or spirit, is some six times that of the petrol cars, the boiler rarely receives the attention it ought to, and water cannot be carried for a longer run than 50 miles at the outside. Electric cars have yet to fight against the heavy odds of the accumulator weight, not less than 15 cwts. of which are required to produce the full voltage.

Petrol cars holding the market so well, despite numerous difficulties in working, a typical car has been illustrated in Fig. 919, of medium size for touring purposes. The early cars were insufficiently powered, and the demand for starting on and racing up average inclines has raised the horse-power very considerably: so we have at present :—

	Carrying	Engine	B.H.P.	Speeds per hour
Voiturettes or smallest cars	2 people	1 cylinder	4 to 8	15 miles
Light Voitures or small cars	3 to 4 people	2 to 3 cylinders	5 to 10	20 ,,
Touring and large cars	4 to 8 people	4 cylinders	12 to 30	30 ,,
Racing cars ...	2 people	4 to 8 cylinders	30 to 75	70 ,,

Heavy traffic must ultimately be one of the largest developments of motor work. At present it is mostly steam-driven, though petrol driving is adopted for omnibuses.

Referring to Fig. 919, the lower part of the car is called the 'chassis,' and the carriage portion the 'carrosserie.' The latter is shewn at *f*, as of 'tonneau' form, with back entrance, which, though compact, is now superseded by the more convenient 'side-entrance' body. Both give two seats in front, one for the driver; and two seats at the back. Often, but not always, there are two frames: the one marked *a*, supporting the carriage and resting on two laminated springs over each axle, carries also the brake and chain pinion shafts; the second, marked *b*, supports the engine, gear case, water tank, &c.; and the two frames are

rigidly connected at front and back, and by extra stays near the engine. The dashboard is at *c*, and the engine bonnet at *k*, the latter pierced with venetian openings for coolness, while *d* is a lubricator leading oil to all main machine parts. The wheels are of 'artillery' pattern, in imitation of those used on gun-carriages, and are generally supplied with pneumatic tyres, pumped to 100 lbs. pressure. The axles *w* and *x* are 'dead,' that is, are prevented from rotation, the wheels merely revolving on the tapered axle ends; but one form of car has a 'live' hind axle driven by bevel gear at its mid-length by a shaft from the gear box having universal joints in its length.

The distribution of weight is such as to put nearly equal loads on front and back wheels, when working; and as the engine and radiator are most conveniently placed in the front, the gear-case and water tank are put in the rear, although this necessitates considerably extra weight in pipes. The petrol tank is at *e*, and the fuel feeds therefrom to the carburettor B, where it is gasified and mixed with air before passing to the cylinders A A by the inlet pipes *h h*. The exhaust pipes then carry the products to the silencer, where they enter at R, pass backward and forward through concentric cylinders, and emerge at S. The engines are usually of the vertical type, with crank chambers underneath, although horizontal types have sometimes been introduced to overcome vibration. The former is, however, most convenient, and if several cylinders are used with pistons travelling in opposite directions, the inertia forces may be fully balanced (see B, Fig. 860, p. 898). The 'tourist' car shewn, of 12 to 15 B.H.P., requires four cylinders, and in addition a large fly-wheel C to equalise crank effort, into which is fitted the friction clutch D. Earlier cars were often belt-driven, which provided a perfect slipping arrangement for obtaining half power and the necessary fine adjustment between change of gears, but belts are much affected by change of weather, and cause much difficulty when riding up hills. The friction clutch is an admitted necessity, and is kept in gear by a strong spring E, giving uniform but not overgrip, which can be released by a foot lever wholly or partially, thus providing the required 'elasticity' of power, only perfectly obtained in the steam engine.

The inlet valves of the engine are automatically opened against springs, though lever-opened valves are now much in favour; and the exhaust valves are lifted by a cam driven by a 1:2 gear from the crank shaft (*see* Fig. 878, p. 915). Ignition has been fully described at p. 963, but a very interesting discovery has been accidentally made by Panhard's workmen: that a break in the secondary circuit, outside the engine, of about 1 $^{m}/_{m}$, will cause the spark to act with greater certainty, even jumping across dirty inside terminals. In smaller un-governed engines, the advance spark (Fig. 898, p. 964) is adopted to regulate the speed by hand, but in the larger engines a centrifugal governor is placed on the second motion or cam shaft, which formerly closed the exhaust valve to lower the speed, but now throttles the mixture. In most cases it is permitted to put the governor out of gear so as to quicken the engine by advance spark or other method. The speeds adopted are higher than for land engines, for this is the only means of reducing engine weight. They vary from 650 to 2000 revs. per m., though few rise to 2000; some makers, running at 650 to 850 with governor, obtaining the higher speeds by hand. Stroke and diameter are often about equal, and thus a simple but rough rule at 1500 revs. is

$$\text{B.H.P.} = \frac{d^3}{10}$$

The divisor 10 becoming 20 at 750 revs., and 15 at 1000 revs.: or for any speed, divisor = 15,000 ÷ revs. If stroke and diameter are not equal, d^2s may be substituted for d^3.

The full revolutions transmitted along shaft G, have to be suitably reduced by change wheels at J in the gear box *g*, giving three forward speeds as required, on shaft H, with a reverse motion through the intermediate wheel K; the changes being effected by the clutch handle L and rod M, which slide the wheels along their shafts, a method which causes much wear, but is simple, and has not been superseded. Thrust bearings are provided at K and V. The motion is carried forward by bevel gear T to the clutch pinion shaft *t*, passing through the compensating or differential gear in box U, already described at p. 526, which serves to disunite chain pinions W W, and permit their

rotation at different speeds when the car is turning a corner. The pinions lastly drive the hind road wheels by chains to the chain wheels X X, the latter being fastened to the spokes by bolt studs. The alternative to chain driving is the substitution of a live axle at *x*, which is adopted in many cars, but the universal joints required in the gear shaft render the method objectionable, for allowance must always be made for deflection of the car springs.

The steering hand wheel at *p* actuates by worm and spur gearing and a lever, the steering rod *r*, which in turn by lever slews the wheels *n* and n_1 round centres 2 and 3, for they are both connected by levers *s s* and rod 4. Levers *s s* are not parallel, so n^1 and n_2 turn through such different arcs that the wheel centre lines will meet in a point coincident with the back centre line *x* produced, all four wheels then truly rolling round one centre in *x*; the principle being known as that of the Ackermann gear. Other movements, such as change speed gear, adjustment of carburettor, governor cut-out, &c., are all worked from handles on steering pillar.

The cylinders are cooled by water kept circulating in jackets by the pump N, and the hot water from cylinders is circulated through the radiator *l* to be cooled, then back to water tank P to be re-conducted to the engine jacket. The brake is provided at Y Y, where two drums are gripped by blocks pulled tight as required, by the lever Z. Formerly, straps were used as at p. 570, but the method being non-reversible, would not prevent the car running backward down-hill. *m* is the engine-starting handle.

The Longuemare Carburettor, illustrated in Figs. 920 and 921, is probably the very best example of a float-feed spray carburettor. In Fig. 920 the petrol feeds a long pipe E, through gauge at F, G being a cleaning cap. A is the float chamber and B the float. The spindle H is weighted at its lower end, and supported by levers D if the float be at lowest position as shewn, so that petrol enters freely. Gradually the float rises until the petrol reaches level C, when the levers D rising with the float permit the weighted needle to fall and close the lower orifice, checking the flow. When in action there is a delicate play of the float between

these limits, and J is a press button for trying if the float be satisfactorily acting. At K the liquid rises through small holes to L, and there is drawn by the suction through fine grooves on the sides of an inverted cone, placed at the level of C, and the petrol enters the chamber M as spray. Air is drawn inward at N, being adjusted in quantity by the circular slide P actuated by lever R, and constrained to pass into M in close contact with the spray. The mixture passes through gauze into the higher chamber, and is permitted to flow to the engine through inlet pipe Q, being further regulated or throttled by the circular valve V, actuated by lever U. A jacket X is supplied with hot exhaust gas through the small pipe W, to perfect the gasification of the mixture.

Fig. 921 shews the arrangement of the carburettor and connections, *h* being the petrol tank and *j* the starting or stopping valve, *g* the flow-pipe to float chamber *a*, *b* the adjusting chamber with levers *f*, to which passes the mixing air through pipe *c* from near the hot engine cylinder *d*, *l* the exhaust from engine to *m* the silencer, a portion of the products being abstracted, by way of cock *p*, along pipe *n* to the warming jacket *b* of the carburettor, and *e* is the inlet pipe to engine with inlet valve at *k*.

P. 768. **Efficiency of Boilers.**—The evaporation obtained in lbs. of water per lb. of coal, the former from and at 212°, is a very ready way of comparing the thermal efficiency of boilers, though giving no indication of other facts. The results following may be taken as having been obtained with best Welsh coal of about 14,500 thermal units calorific value per lb. of fuel :—

	Evaporation.
Ordinary Lancashire boilers	10 lbs.
Galloway breeches boiler	12·8 ,,
Babcock water-tube boiler	12·6 ,,
Climax water-tube boiler	12·2 ,,
Thornycroft water-tube boiler	11·7 ,,
Belleville water-tube boiler	11 ,,
Cochrane vertical boiler	9·1 ,,

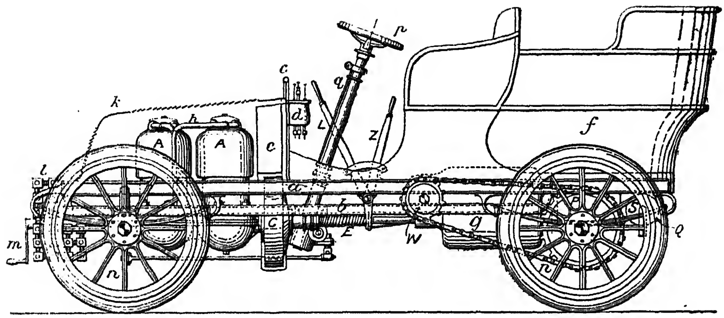
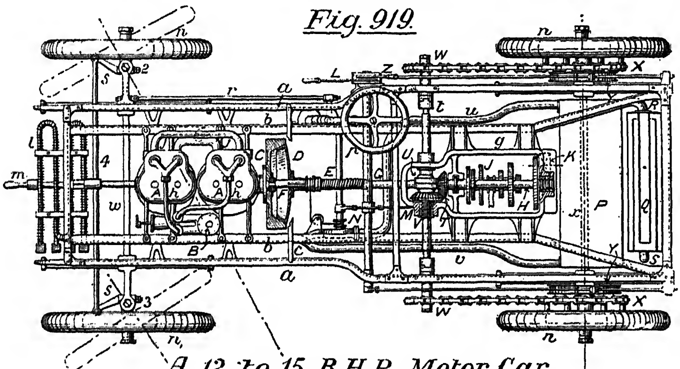


Fig. 919.



A 12 to 15 B.H.P. Motor Car.

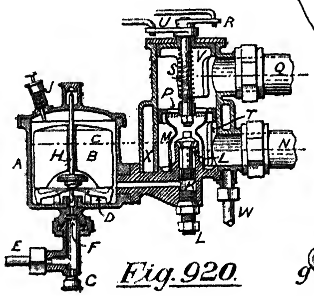


Fig. 920.

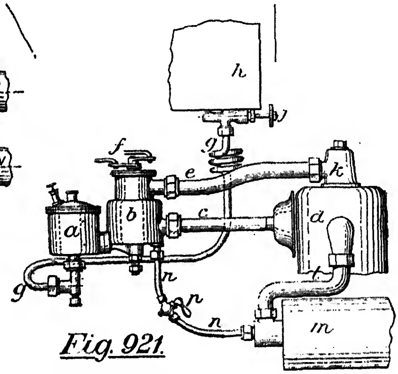


Fig. 921.

The Longuemare Carburettor.

P. 901. **Water Softening.**—It is now generally admitted that the proper place to soften hard feed water is in the feed tank, and not inside the boiler. The chemicals to be added are dealt with at p. 901, and it remains to describe a feed-water softening apparatus for external use. The 'Bruun-Lowener' softener in Fig. 918 is remarkable for its simplicity and small size. The hard water enters at pipe A, being checked as required by ball valve B, passing by pipe C C into the cataract vessel D, which is divided by a mid-partition, so that when one side is filling the other is emptying. At each discharge of D three things have been done: the cam roller T has passed under the spindle R, raising the lever U on centre V, and with it the valve rod W, thus admitting a certain quantity of liquid chemical from tank Q into D; the paddle S has been automatically moved within Q to keep the chemicals mixed; and the paddle P has travelled across compartment E so as to stir the discharged chemical among the feed water. The treated water next rises in chamber Y to flow down the vertical tube F, but on its way is met by a jet of steam through pipe M and distributing nozzle N, which has the effect of accelerating the deposition of precipitates, thus reducing the necessary size of the apparatus. Continuing, the water passes through filter H and emerges into chamber J, whence it is pumped through pipe K to the boiler. Mud is withdrawn through cock L. The apparatus shewn is large enough to supply 400 gallons of water per hour.

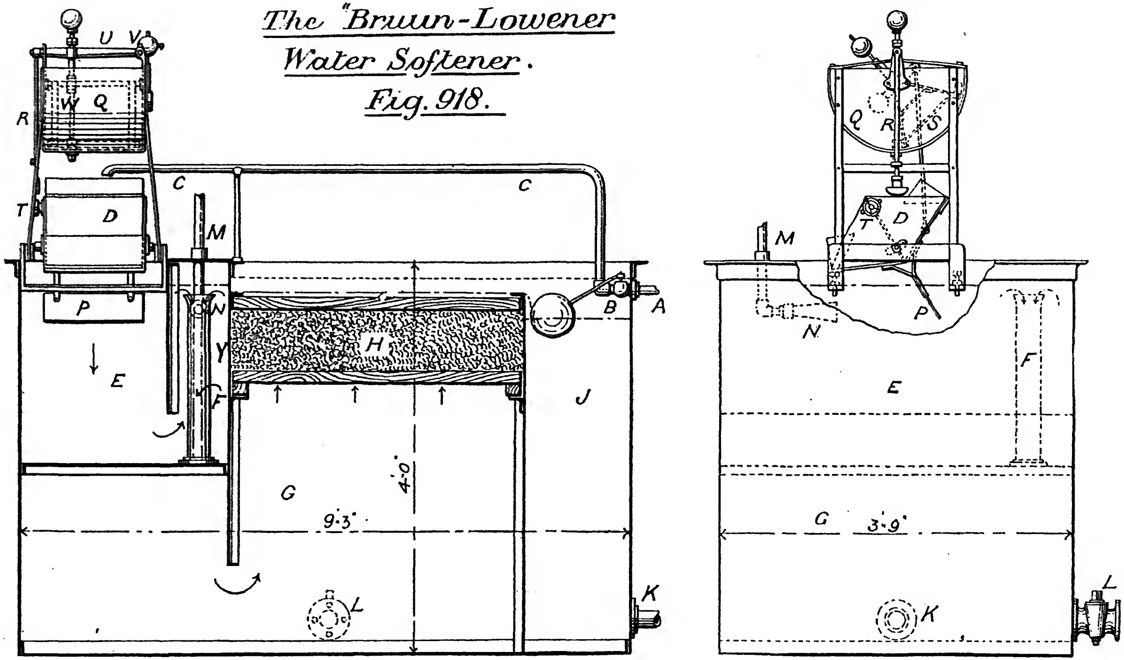
CHAPTER XI.

P. 710. **Density of Fresh Water.**—In Fig. 922 a curve has been prepared to shew accurately the variation in the density of fresh water between freezing and boiling points. The number of pounds weight per cubic foot may be read off by means of the scale on the left of the diagram by measuring the height of the curve above the base of 60 pounds, referring to the scale, and adding to it the figure 60.

P. 738. **Balanced Hydraulic Lift.**—Mr. Ellington's lift described on p. 738 is no longer manufactured on account of the general objection to internal packings. The Balanced Hydraulic

The "Bruun-Lowener Water Softener.

Fig. 918.



Passenger Lift shewn in Fig. 923, designed and manufactured by Messrs. Waygood and Otis, is a most satisfactory modern apparatus, dispensing with all internal packing by the substitution of rams for pistons. The central picture is an enlarged section of the working cylinder A and the rams, of which C is the displacement ram, lying within the cylinder and riding vertically upon the power ram D, the last being held rigidly by bolts E E and cross-head F. Pressure is admitted to space G, and only displacement

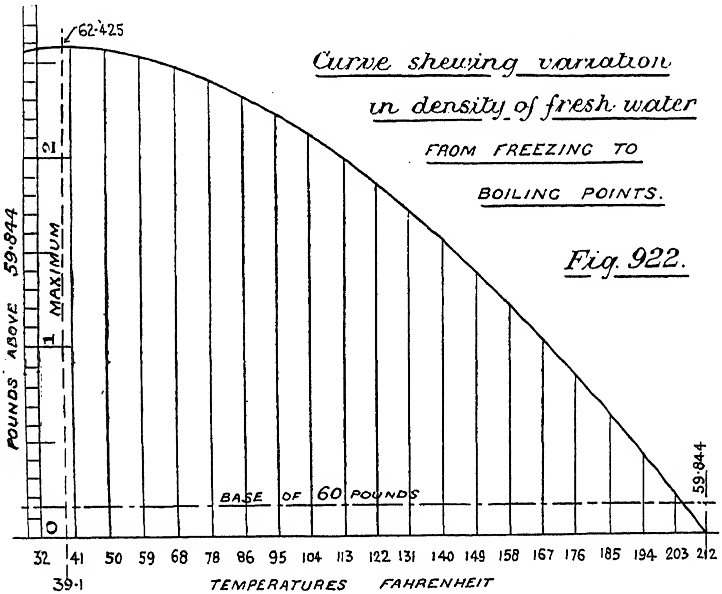


Fig. 922.

water lies between A and C. Upon the latter is always a minimum pressure due to weights H, which balance the greater portion of the lift ram and cage, the unbalanced weight being necessary on the cage side to bring the lift down when water in G is opened to exhaust. The buffers J J are a precaution, but are not rested upon as a rule.

Referring now to the general drawing: the cage K rises from the basement to a third floor in the example given, or a total of 52 feet. The controlling valve L is worked from the cord N

Hydraulic Balanced Passenger Lift.

BY MESSRS. WAYGOOD & OTIS

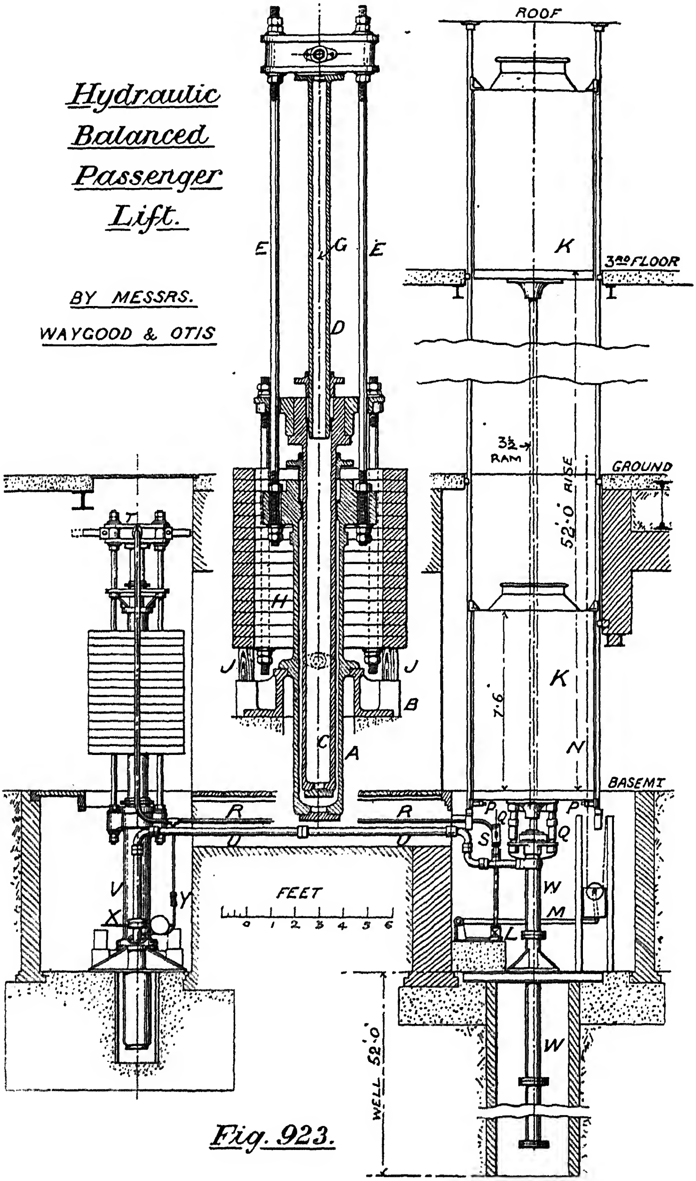


Fig. 923.

through lever M, but when the cage reaches its highest or lowest position the pressure water is cut off automatically at valve L by the striking of tappets such as P P, and the load is received on buffers Q Q when at lowest position. The pipe R transmits pressure water to the power cylinder, entering at T; and the pipe U U connects the displacement cylinder V with the lift cylinder W. As mentioned, the pressure of the displacement water will depend upon the load, which varies with the passengers, the minimum being somewhat under the weight of lift ram and cage. A relief valve S is placed upon pipe U, and as the water in U gets lost by leakage and the weights H fall too low, the valve X is automatically opened, admitting pressure water from R to cylinder V through pipe Y, wherein is a non-return valve.

APPENDIX VI.

(EIGHTH EDITION.)

CHAPTER I.

P. 3. **The Cupola.**—A thoroughly modern foundry cupola is shewn in Fig. 924. It is known as Stewart's patent 'Rapid' Cupola, and is made by Thwaites Bros. of Bradford, its principal

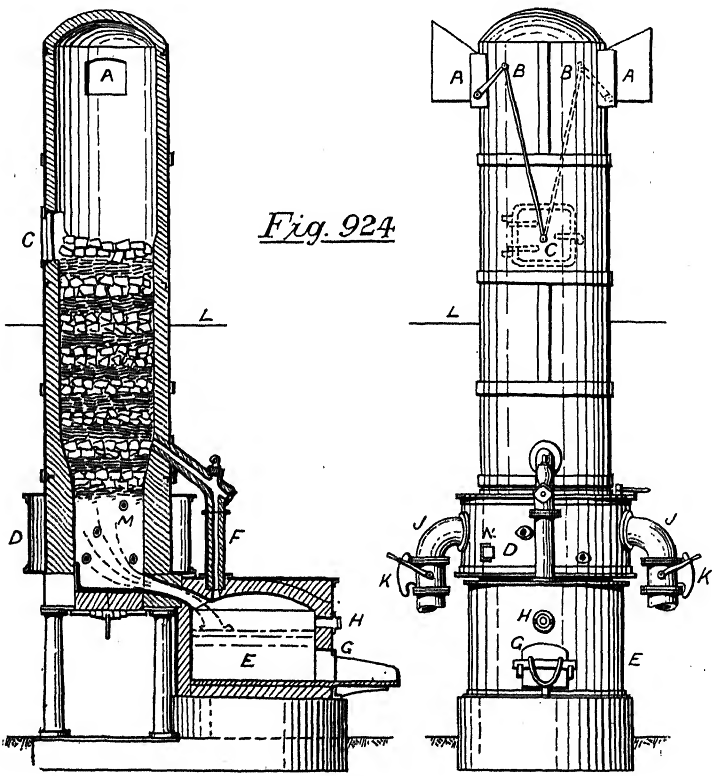


Fig. 924

Stewart's "Rapid" Cupola.

features being the blast box D and the reservoir E for the molten metal. The former is now applied to all new cupolas on account of its quiet and steady action, the warmed air passing in at several openings such as M. The reservoir keeps the metal quite hot until required, and there is the usual tapping door G, as well as an air pipe F lined with ganister, and a sight hole H. The charging door C and the damper rods B B are worked from the charging platform L, and the gases escape at A A. The blast pipes J J are governed by valves K K, and there is another sight hole at N.

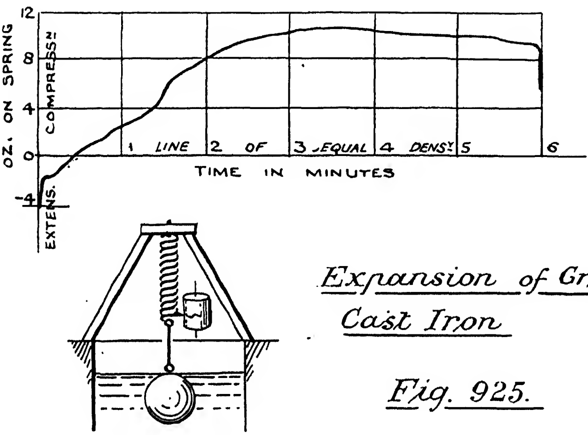
Pp. 6 and 747. **Blackening.**—Plumbago is now very much used for the facing of moulds, being dusted on like the charcoal powder, and afterwards sleeked with suitable tools. It must be used sparingly owing to its refractory nature, and the consequent tendency to close the sand pores and prevent the escape of gas. Other substances have also been adopted, but plumbago is the best because it lies firmly on the mould during casting, while oak-charcoal dust, though otherwise excellent, has a tendency to float.

CHAPTER II.

P. 71. **Expansion of Cast Iron** from the solid to the liquid state.

An interesting experiment was made by Sir Thos. Wrightson. He placed a cast-iron ball of 132 oz. weight on a surface of molten cast iron. At first the ball sank below the surface, which shewed it had a greater density than that of the liquid; but in a few seconds it had just risen level with the surface, having then an equal density with the liquid. Continuing to rise, about $\frac{1}{16}$ of its volume shewed in 3 or 4 minutes, and at the end of 6 minutes it had melted away. The top of the ball was connected with a spring balance which carried a pencil that marked the curves shewn in Fig. 925 upon a clock-driven drum. Comparing the pressures on the spring, it appeared that the ball expanded 1% before it attained an equal density with the liquid, and a [illegible]her expansion of 6% caused a total increase of 7% in volume.

The first rise of 1% would correspond to the 'contraction allowance' of the pattern maker.



Expansion of Grey Cast Iron

Fig. 925.

CHAPTER III.

P. 76. **Wrought Iron.**—The following notes are given on the authority of Mr. James P. Roe, of America, inventor of the Roe puddling process:—

Puddling permits the use of pigs with very various composition, up to 3% Si, 3% P, 2½% Mn, or ·35% S, though not all in the same pigs; but the best proportions are 1% Si, less than 1% P, ·1% S, and ·5% Mn. The wrought iron produced is free from red or cold shortness, and is very weldable. The flux being roll-scale, ore, or oxidising gas, is for the purpose of abstracting the C as already described, and forms the 'cinder,' which when low produces the greatest strength, but when high gives very free welding power. The subsequent piling and reheating is for the purpose of removing this cinder to the desired extent.

In the final structure of the metal each grain or crystal is surrounded by an envelope of cinder (ferrous silicate), the greater part of which, however, is eliminated by hammering and rolling, and a fibrous structure results, which is the real source of strength

in wrought iron, so that screw threads will withstand fracture when steel threads break at the root. The elongation is more uniform than that of steel, though of less extent; and the cinder, as already explained, facilitates welding. Also wrought iron oxidises less freely than steel, where the manganese is found to cause pitting.

The principal defect of wrought iron is its transverse weakness, though this is reduced by the elimination of cinder; while the real practical objections are its higher cost, and the difficulty of procuring labour for the puddling.

P. 800. **Aluminium** shrinks largely in cooling, and is not poured therefore directly into the mould, but into what is called a runner box, from which it flows into the mould when certain iron plugs are withdrawn. The runners and risers must be ample in size, and there should be plenty of vents. The blackening consists of graphite and water. The metal is melted in crucibles, at a temperature of 1210° F.; and most of the alloys are treated in the same manner.

As pure aluminium is soft and weak, it is always alloyed where strength is required. Wolframinium and Romanium are 'light' alloys having up to 10% of other metals, of which the principal is tungsten, together with some copper and nickel. Of 'heavy' alloys, R_3 breaks at 34 tons per sq. in., and has 23% elongation in 2″, while R_4 has 42 tons breaking and 10% elongation.

Aluminium is rolled cold into very thin plates, but must be afterwards annealed, and may be forged at a low heat. It must only be filed with single-cut files to avoid clogging, and must be cut at high speed with a lubricant of turpentine or petroleum. Its uses are various, but it is mostly adopted where light weight is an advantage. It has been tried for the hulls of launches, but not with success, because of its rapid corrosion in salt water. Alloyed with 6% of copper, however, it is not more corrodible than iron or steel. When added to steel ingots during melting, in the proportion of 1 to 1000, it prevents piping and porousness.

CHAPTER IV.

P. 100. **The Power Hammer,** so called because it is actuated by mechanical power, rather than by steam or other gas, is of considerable service for lighter forging in smiths' shops, being made in various sizes from 1 lb. to 5 cwt., but rarely beyond the latter. As originally constructed (*see* 33, Fig. 317, p. 349), the rotating shaft rocked a lever whose fulcrum could be varied to alter the stroke and power of the blow; but the present form depends entirely upon the adjustment of an air cushion, the stroke of the lever being invariable.

The Longworth power-hammer as manufactured by Messrs. Samuelson & Co., Figs. 926–7–8, will serve to illustrate the new departure. Fig. 926 is a general view, and Fig. 927 a section through the cylinders; and the same lettering is adopted throughout. Fast and loose pullies are supplied at N, the driving pulley being also a flywheel, to secure uniformity of speed; and U is the striking gear for moving the belt fork V. The rotation of the small crank P causes a connecting rod Q to vibrate the bent lever R, which is further connected to the upper or 'actuating' cylinder by the coupling rods S S. This cylinder is supplied with air holes at D and E; and the piston A, together with the rod A C and the tup, is therefore secondarily moved up and down through the means of an air cushion contained in D. On the upstroke the action is easily understood, but on the downstroke it must be explained that the cylinder D travels at a higher rate than that of the freely falling tup, and gaining upon it, causes a sharper blow than would otherwise occur. The lower and fixed cylinder H is for the purpose of regulating the blow, or of preventing it altogether; and this is effected by the valve K, which can be made to cover any number of the ports at J, or to entirely open them. This valve is worked by the attendant, either by hand at L or by the foot lever W and rod M, the release to the normal position of the diagram (for complete inaction) being given by the long spring at M.

The working will now be easily understood. Putting the belt upon the flywheel, the cylinder commences to move up and down at 260 strokes per minute, but the tup never reaches the

anvil because the ports at J are all closed, and the air cushion under piston B prevents the blow. Moving K downwards, the air underneath B is wholly or partially released, and the tup therefore strikes the anvil with whatever force may be desired, while a rapid opening and closing of K will effect a single blow. The valve T automatically opens to admit air on the upstroke, and the upper part of cylinder H is always open to the atmosphere.

Fig. 928 is a diagram of velocities, where the ordinates V_C indicate the uniform velocity of the crank pin of 5·88 ft. per sec., and the radial ordinates V_L within the curve shew the velocity of the lever pin for any given position in the vibration. It will be seen that the latter becomes 9·5 ft. per sec. on the downstroke, or much greater than the velocity of the crank pin, and proves the smartness of the blow as conferred by the mechanism.

P. 100. **The Drop Hammer,** Fig. 929, is now much used for stamping purposes, and the blow is caused entirely by the action of gravity on the tup. The bed T carries two side standards B B, and the anvil N; and the tup C is free to fall

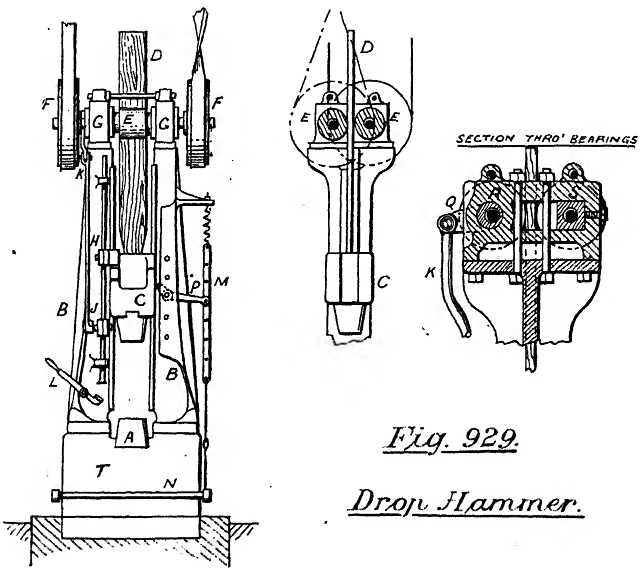


Fig. 929.

Drop Hammer.

3 CWT. SIZE

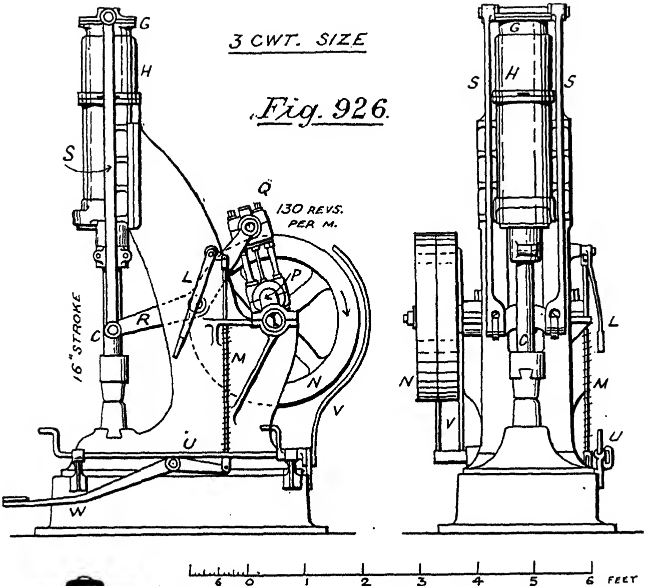


Fig. 926.

The Longworth Power-Hammer

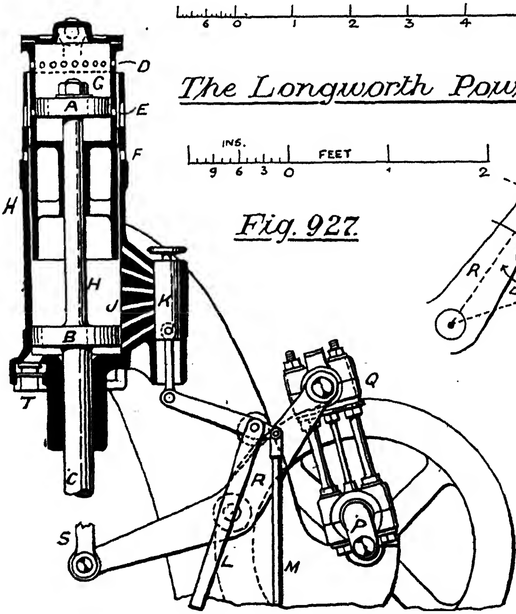


Fig. 927.

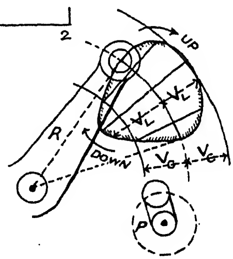
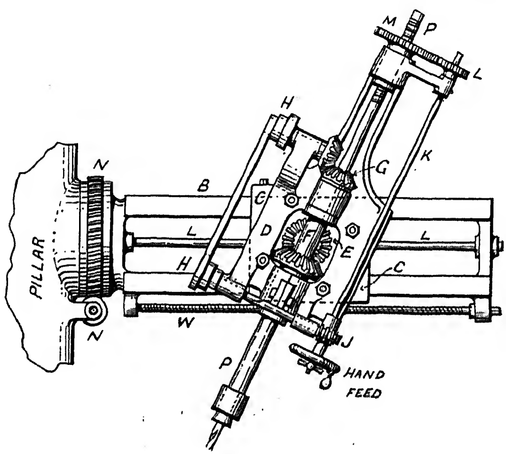


Fig. 928.

between the guides so formed, being lifted by the board D, which passes between the friction rollers EE. The rollers are driven by separate pullies F F, one with a crossed and the other with an open strap, the rotation being always such as to cause the tup-board to rise. The brackets G G contain the bearings, the back pair S being fixed, while the front form eccentric bearings that can be rotated by the lever Q so as to release the frictional grip between the rollers and the board. The releasing gear is automatic, and consists of an adjustable tappet H, which is caught by the tup on the upstroke, thus causing the rod J K to be lifted, and the lever Q rotated so far as to permit the tup and its board to fall. The hand lever L is for the same purpose, and is principally convenient for plain forging, where a succession of light blows may be required. To keep the tup in a raised position, the catch lever P is supplied, and the depression of rod M by the foot lever N permits the tup to again fall freely.

CHAPTER V.

P. 166. **Universal Radial Drill.**—The description of the usual form of radial drill at p. 166 will serve as an introduction to the universal form shewn in Fig. 930. In the latter



Universal Radial Drill. Fig. 930

the drill spindle P can be placed at any desired angle, which is effected by allowing the horizontal slide B to be turned on its long axis by the worm gear N, while the front saddle D is rotated in the plane of the paper upon a back saddle C. The saddles are traversed by the screw W to the required position, and the drive is through shaft L and mitre gear E to the sleeve F, and thence to the spindle P. The smaller mitre gear G actuates the feed, through belt cones H H, worm gear at S, and spur wheels L M; hand feed being obtainable in the usual manner by shaft K to the spur gear. The whole constitutes an exceedingly convenient design of radial drill.

P. 168. **The Square-hole Drilling Machine** (Ainley–Oakes) is a very interesting and remarkable contrivance, for by its aid true and sharp square or hexagonal holes may be actually drilled with accuracy. Referring to Fig. 931, a vertical standard carries a bracketted table X, which may be raised to any convenient position by the screw Y and then fixed. Upon this table are cross slides for adjustment of the work, and a worm gear W for rotation of the vertical axis of the work table V to any desired angle. The drill spindle G carries a specially formed tool F, and is encased in, and driven by, the sleeve E through the mitre gear C on the outer sleeve D, A being the main driving cone, and B the driving shaft Upon the sleeve E are formed suitable cams, shewn in section at *a a*, which are made to engage within fixed cam plates at T T when required, such engagement or disengagement being effected by a lowering or raising of the sleeve E by means of the lever P K and the parallel motion H J, after which P is fixed on its section plate. The side motion or wobble caused by the engagement of the sleeve E with the cam plates is allowed for in the coupling U, which connects the spindle to the feed screw M, and the feed is obtained in the usual manner by power through belt gear S, worm gear Q, spindle P, and spur gear N to the screw M; or by hand wheel R.

The cam plates for square holes are shewn at Z, and three different sizes at least are supplied, while round holes may be drilled by the aid of the plates Z_1; any pair of plates being put in or out of gear by means of right- and left-handed screws that

separate or approach the half-plates. The form of tool for square holes is given at F, and the corresponding cam on the sleeve at E, the rotation of E with the plates causing the tool to successively enter the corners of the dotted square, which is the hole thereby formed in the work.

P. 175. **Further Milling Examples** are given in Fig. 932. At A is shewn a 'gang' of milling cutters fixed upon the milling spindle, by which work of very intricate section may be very rapidly performed, though considerable driving power is naturally required. The bolt slots at C and D could scarcely be tooled with speed and accuracy by any other method than that of milling, and E shews a use for the face cutter. The plane table at B is being machined both on top and sides at one cut, the sides by means of face cutters, and the top by two long cutters having interrupted teeth, thus giving a clearance for the shavings which is much needed in roughing cuts. The rig at H is required when milling a keyway in a round shaft, and consists of two side clamps, further accuracy of position being secured by the dummy key in the lower keyway that has already been cut. A twist drill is being formed at G, where the table is swivelled on its vertical axis to the correct angle, and the drill is rotated in a bearing at the proper rate of speed to mill the helical groove, the method being that of screw-cutting with a mill as the tool.

A grooved cam plate of any required design may be milled by the apparatus shewn at F, which is a plan view. An auxiliary table *a* is fixed to the machine table *l*; and a slide *b* carries a spindle *h*, upon which is the copy *c* and the work *g*. When in operation the weight *e* pulls the slide *b* till it rests against the dummy *d* fixed on *a*, and a very slow rotation of the spindle, or feed, is caused by the worm gear *j*, separately driven by the cone pulley *k*. The result is that the mill *f* forms a groove in the work that is the exact counterpart of the copy *c*.

Pp. 180 and 811. **Milling Machine with Universal Head.**—Fig. 933 shews a large milling machine that may be easily understood from previous descriptions. The bed A has a horizontal slide, upon which is the saddle and cross slide M, and the table B rotated by worm gear N: any desired ratio of feed

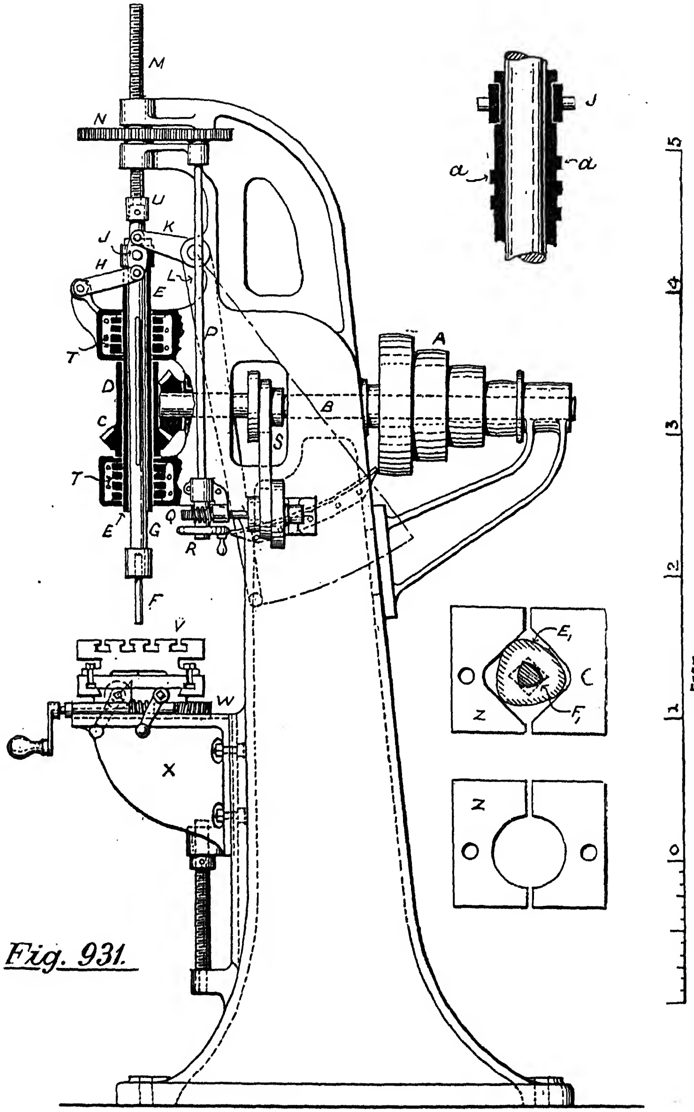


Fig. 931.

The Square-hole Drilling Machine.

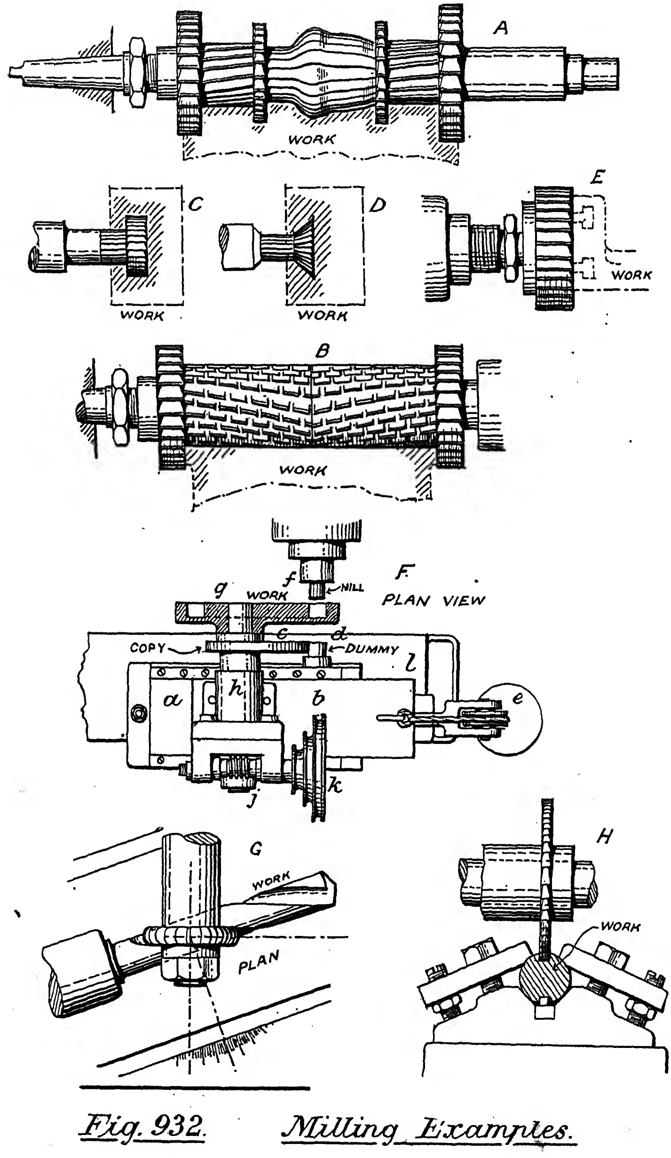


Fig. 932. Milling Examples.

ained by change wheels L; and the feed shaft J is driven gear K in either direction at will. The driving cone C, gear, rotates the tool shaft Q through mitre gear T, haft, spur gear D, and shaft E; and a vertical feed is ined when required through the screw P and change H. The saddle R carries the milling head, and moves

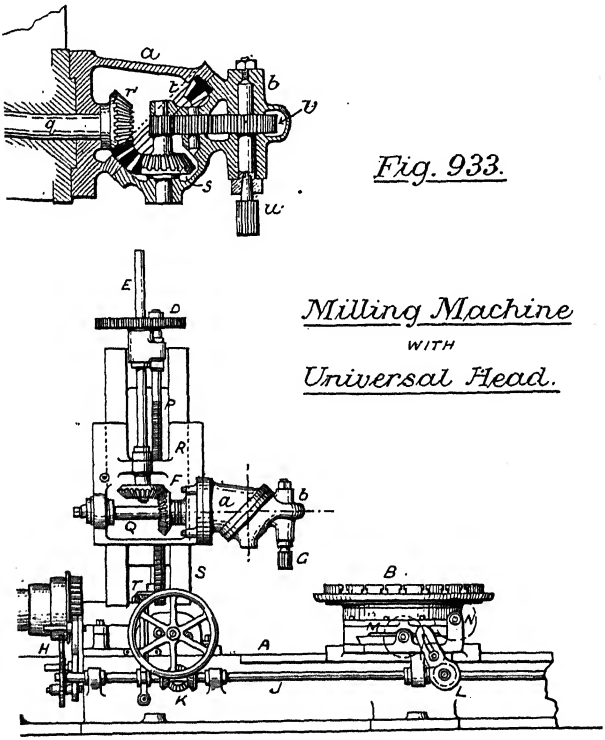
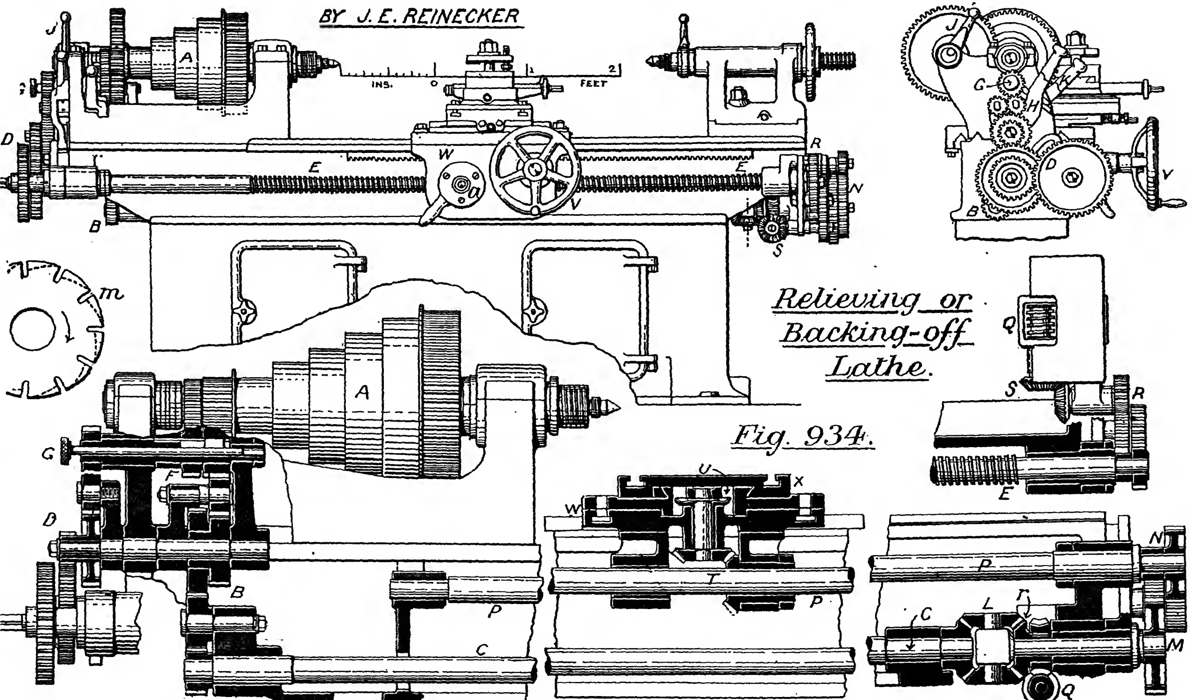


Fig. 933.

Milling Machine
WITH
Universal Head.

vertical slide S. The universal head consists of two d *b*, the former of which can be rotated round a axis to any suitable position, and be there fixed by part *b* may similarly be rotated, but on an axis 45° to that of *a*, and in consequence the tool axis *b u* ced in any position whatsoever. To drive the tool

BY J. E. REINECKER



Relieving or Backing-off Lathe.

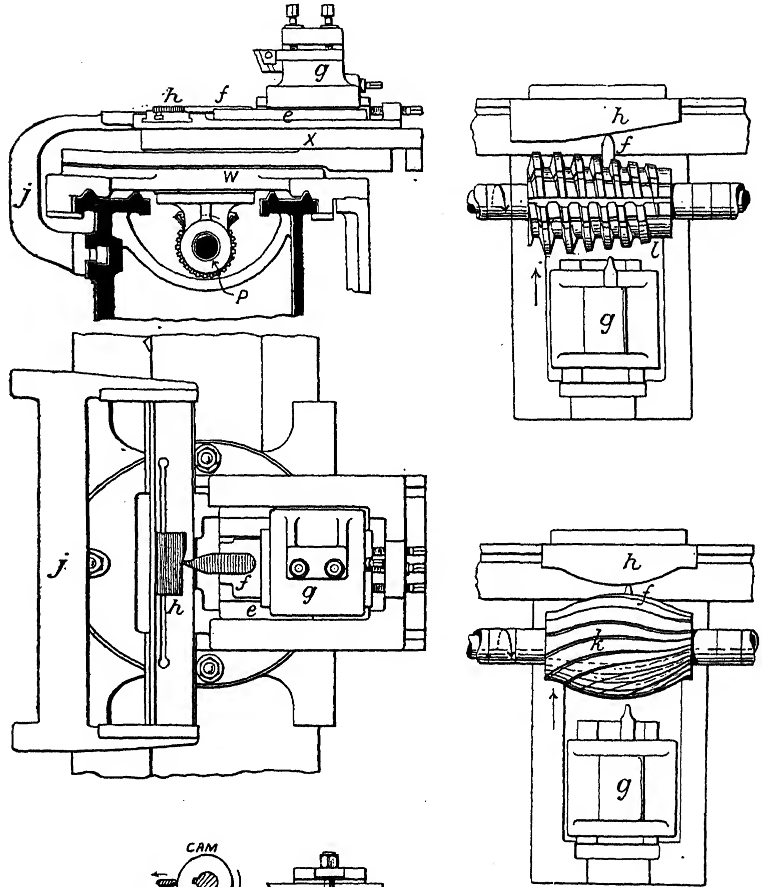
Fig. 934.

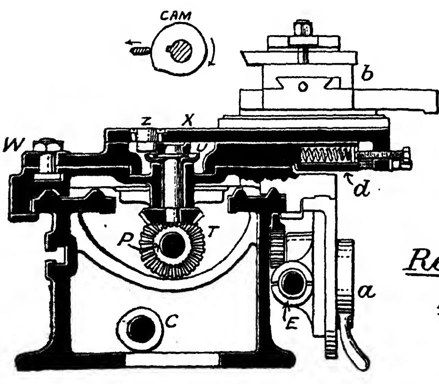
under all circumstances, mitre wheels *r* and *s* engage through the medium of a face wheel *l*, and the motion is transmitted to the tool spindle through the spur gear *v*. The interposition of the face wheel permits therefore a connection of *r* and *s*, whatever the angular relation of *a* and *b*.

CHAPTER VI.

P. 193. **Relieving or Backing-off Lathe**:—In the description of the making of screw-taps, p. 192, the clearance or backing-off was said to be done by hand, but this is by no means the best or proper method of doing it. Taps are now relieved or backed-off by the aid of some machine appliance, which gives the turning or screwing tool such a transverse motion as will cause it to cut deeper into the tap at certain times, as shewn in section at Fig. 203, p. 194, the flutes having been previously cut as at A, Fig. 185. Relieving lathes, specially built for backing-off while turning, have a much wider and more important field of application, however, in the making of milling cutters of every conceivable form, and the machine shewn in Figs. 934, 934*a*, and 934*b*, built by Mr. J. E. Reinecker, and introduced in England by Messrs. Pfeil & Co., from whom the drawings have been obtained, will now be described.

The general view of the lathe is given in Fig. 934, the lower diagram being a section through the bed, so as to shew the arrangement of the shafts. The driving is as usual by cone pulley A and back gear, and the milling cutter to be formed is mounted on a bar and placed between the cone centres so as to rotate at a suitable speed for the cutting. The cutter blank has been previously roughed out in the manner shewn at *m*, where the deep radial grooves indicate the number of teeth to be adopted. These grooves are generally of spiral form, having long pitch, so can easily be done in the ordinary milling machine between centres that are slowly rotated by what is called a spiral attachment. The object of the relieving lathe is next to give each tooth of the cutter *m* its proper clearance angle by a process of turning, the material being removed down to the dotted lines.





<u>Relieving Lathe</u>

<u>BY J. E. REINECKER.</u>

<u>Fig. 934 a.</u>

The direction of rotation being shewn by the arrow, it is evident that the tool must be gradually pushed farther into the cut for each tooth, and then suddenly withdrawn to begin upon the next tooth; while, at the same time, a longitudinal traverse of either a parallel or spiral form is to be maintained as required. We shall now examine the motions of the lathe to see how the aforesaid operations may be performed.

The leading screw E provides the longitudinal feed or traverse, being driven by change wheels at D, and having the usual rocker lever H to reverse the motion when required (*see* Figs. 135–6, p. 145). The spiral shaft C, as it is called, is driven by spur wheels at B, a change of speed being obtained by the sliding spindle G, which puts one or other of the wheels at F in mesh at will; and a rocker lever is also provided at K for reversal. Following shaft C rightwards, it is seen to drive a shaft M through the medium of differential bevels, so that if wheel *r* be fixed, the pinions L roll round and rotate M at half the speed of C. Change wheels N are then so arranged as to turn shaft P, called the cam shaft, at the speed required to cause a prescribed number of teeth to be formed upon the cutter blank; in other words, the rotations of P are always an aliquot number of parts of those of M, and at present also of C, though the latter will only hold when *r* is stationary and parallel-toothed mills are being cut. The cam shaft P is so called because it rotates the backing-off cam U through the bevel wheels T and a vertical shaft, giving the required transverse motion to the upper slide X of the saddle W.

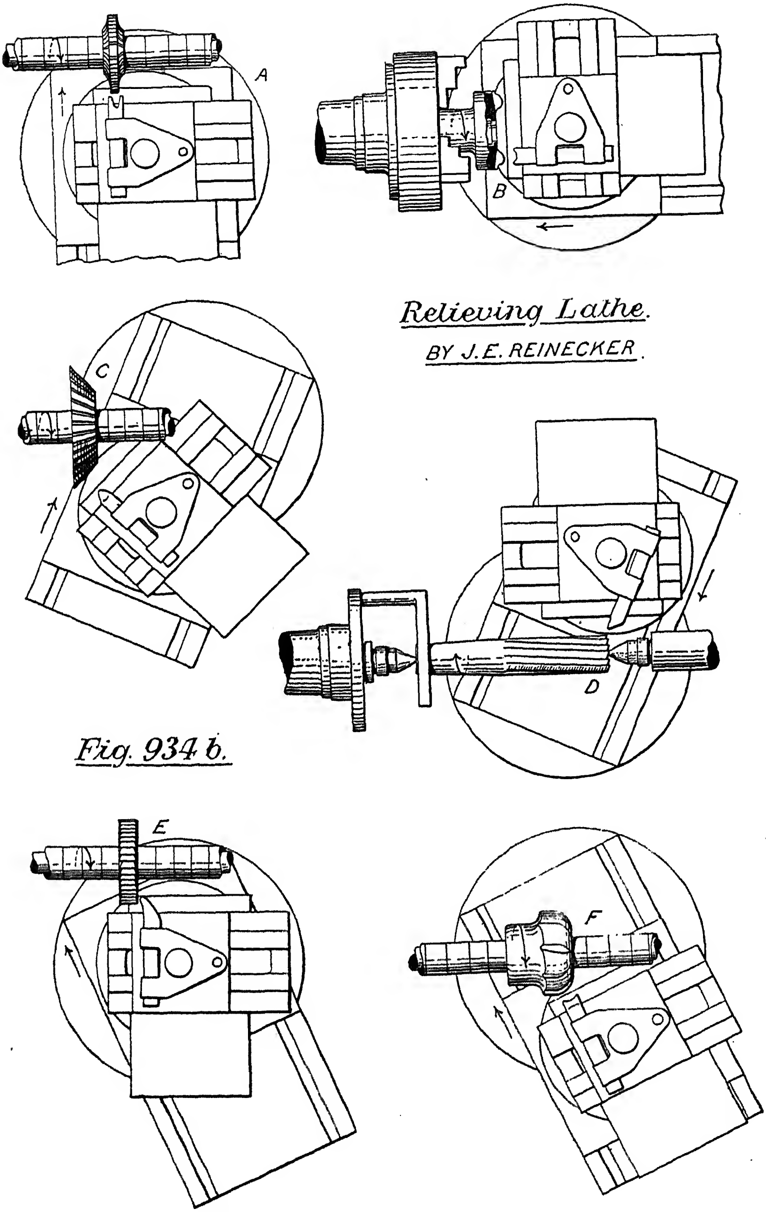
The longitudinal feed is made spiral by means of a curious connection between the leading screw and the spiral shaft. The screw E, supplied with change wheels R at its right-hand end, drives a worm gear Q through the mitre wheels S, thus causing a very slow rotation of the wheel *r*. The shaft M no longer turns at the exact half-speed of M, nor do P's rotations divide exactly into those of C, though the old relationship is retained between P & M. The effect is to cause the reciprocation of the tool slide X, as given by the cam U, to be very slightly retarded, so that at every rotation of the work the tool enters at a somewhat later or lower position. Simultaneously a leftward feed has been given by the leading screw, and thus a left-hand spiral path is traced upon the

blank by the tool point. A rack traverse V is provided for setting, and a nut clamp at *a* for the leading screw.

The lower diagram of Fig. 934*a* is a cross-section through the saddle W, shewing the shaft P and bevels T rotating the cam U, the shape of which is seen above. The spring *d* within the saddle causes slide X to bear upon the cam through the stop Z, and as the milling cutter slowly revolves with the mandrel, the tool in rest *b* commences to cut on the circumference of the blank, sinks deeper, and emerges suddenly for a second tooth, as previously described. The traverse can be at times prevented while all the other motions are retained, by releasing the half-nuts at E, a method required when forming spur-wheel and such-like cutters as at Fig. 186, p. 178. In fact, that diagram and the attached description shew clearly the advantage of machine-relieved milling cutters, where the relief angle is the same for any radius; and it follows that if only the front of the tooth be ground by a suitable emery wheel, held with its flat surface as a radius of the cutter, the form of tooth is invariable so long as the cutter lasts. The object of Reinecker's lathe is to make this condition for *all* milling cutters and not merely for those in Fig. 186.

Various cutters in process of turning and relieving are given in Fig. 934*b*. That at A is a spur-wheel cutter where no traverse is required. The direction of rotation and cam reciprocation is shewn in every case by arrows. Thus at B the top slide is turned round through 90°, and its reciprocations are in a longitudinal direction, which are the motions required for the special face-cutter shown; and again the traverse is put out of action. Other face-cutters are being tooled at D and E, while the special mills at C and F require no further explanation.

Referring again to Fig. 934*a*, the upper diagrams illustrate the copying attachment required for the cutting of solids of revolution other than simple cylinders. A long bracket *j* is fixed firmly to the back of the bed, and carries a rigid copy *h*, representing the profile of the required cutter. The saddle W carries the reciprocating slide X as usual for the formation of the cutter teeth by relief motion; but the tool box is placed on a second slide *e* on the top of X, which is pressed forward by a spring so as to cause the pointer *f* to rest on the copy *h*, thus transferring the profile to



Relieving Lathe.

BY J. E. REINECKER.

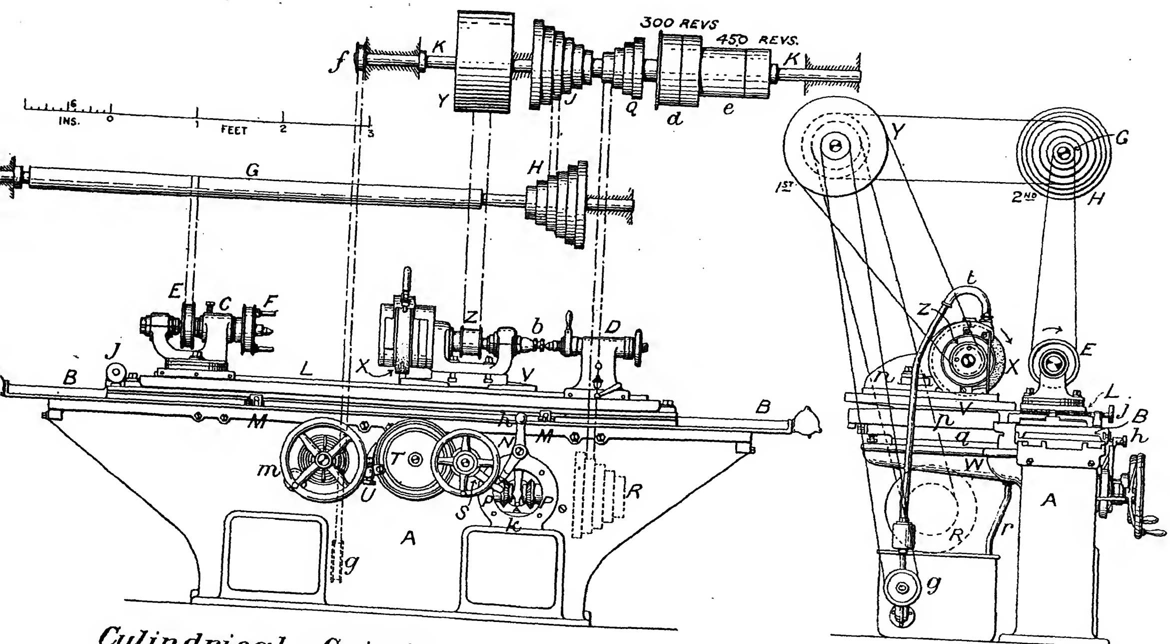
Fig. 934 b.

the cutter circumference. The examples *k* and *l* serve as illustrations. The worm-hob *l* is tapered to suit a wheel-cutting system of Reinecker's own, where the centres of worm and wheel are never changed during cutting, the worm being simply advanced along its axis to obtain the depth feed, and thus the correct shape of the tooth space is obtained right through the operation.

Lastly, it may be said that the lathe just described will take work up to $9\frac{3}{4}$ ins. diameter and 39 ins. long, and will relieve cutters of from 2 to 40 teeth, having spirals of from 4 to 400 ins. pitch, right or left hand. The 'hand' of the spiral is entirely determined by the change wheel R, Fig. 934, for if the wheel *r* turns slowly in one direction the tool lags, but if in the opposite direction it gains, on the work. Also the number of teeth in the blank is determined by the change wheels N, the traverse by those at D, and the depth of cut by a hand advance.

P. 197. **Machine Grinding.**—The Emery grinding machines on pp. 198–9, it will be observed, have their operations confined to the accurate sharpening of cutting tools, especially of milling cutters, twist drills, rimers, and the like. It was soon found that grinding by emery wheel was an exceedingly satisfactory means of finishing all kinds of machined work requiring extreme accuracy. Ordinary tooled work always required correction after machining because of the inaccuracy produced by clamping or by the pressure of the tool, and this finishing process was formerly done by hand with file and scraper in the case of flat surfaces, or by lapping with emery in that of lathe or bored work, the object being to remove the metal by means of light cuts with little stress. Lapping consisted of the application of a leather-faced tool or disc of leather soaked in oil and supplied with emery powder. It is not so accurate as might be wished because of the tendency to wear down soft spots more than the hard parts of the work, and although the principle is still followed for the last finish, a tool of soft cast iron is preferred, fed with crocus powder and oil. The old methods would have remained but for the gradual and sure advance in the perfection of manufacture of the emery wheel.

Back in 1877, or thereabouts, when these wheels were being first introduced, the binding material of the emery was so unsatisfactory as to cause glazing by not falling away with the wear of the emery particles, and the stones often broke by centrifugal force due to insufficient strength in the material. The wheels of the present day are a great advance on these former ones, and some notion of this improvement may be gained when it is stated that $\frac{1}{16}$ in. thickness of wheel may be adopted without breaking. All emery wheels were formerly made from corundum or from emery powder, or both (amorphous conditions of the sapphire and ruby), but some very valuable artificial products are now on the market in addition to the natural substances. Carborundum is one of these, consisting of salt, sawdust, and coke subjected to great heat in electric furnaces for 24 hours, the crystals produced being ground and sifted. For use it is mixed with kaolin and felspar, and then baked into wheels. This material is suited for grinding hard cast iron and mild steel, but for hardened steel the older emery appears to be the best, the latter being pressed into moulds and baked with some binding material that is itself abrasive, and will therefore wear away at the same rate as the grinding material. The harder the work to be ground, the coarser and softer the wheel should be, and the surface speed should be about 6000 ft. per min. for mild steel, 4000 to 5000 for hard steel, and 4000 for cast iron. It is most important that there should be a plentiful supply of cold water (not soda) to the grinding point, which is to prevent the softening action due to the heat caused in grinding, and it is also unwise to use the same water over and over again, as it becomes charged with grease and glazes the wheel. A greater truth and better finish also results in the use of water, but the best possible effect is produced when a little lapping is finally done on the job, thus removing the grinding marks and giving great accuracy. If water cannot be fed to the wheel, as is sometimes the case with internal grinding, it should be poured on the outside of the work. All long work should be well steadied, or untruth will certainly occur through springing, and a roughing down to approximate size should always precede the actual grinding, so as to leave as little as possible for the emery wheel to do. It must be realised that although a very



Cylindrical Grinding Machine

BY J. E. REINECKER

Fig. 935.

light cut is taken by the wheel, work can be spoilt by over-grinding, and a very small advance of cutting depth together with a rapid traverse is the best means of securing accuracy. At the same time the work should be well watched and continuously gauged. Lastly, the machines should be constructed so as to move the work under the wheel in the opposite direction to the rotation of the latter, and with these preliminaries we will now examine the grinding machines themselves.

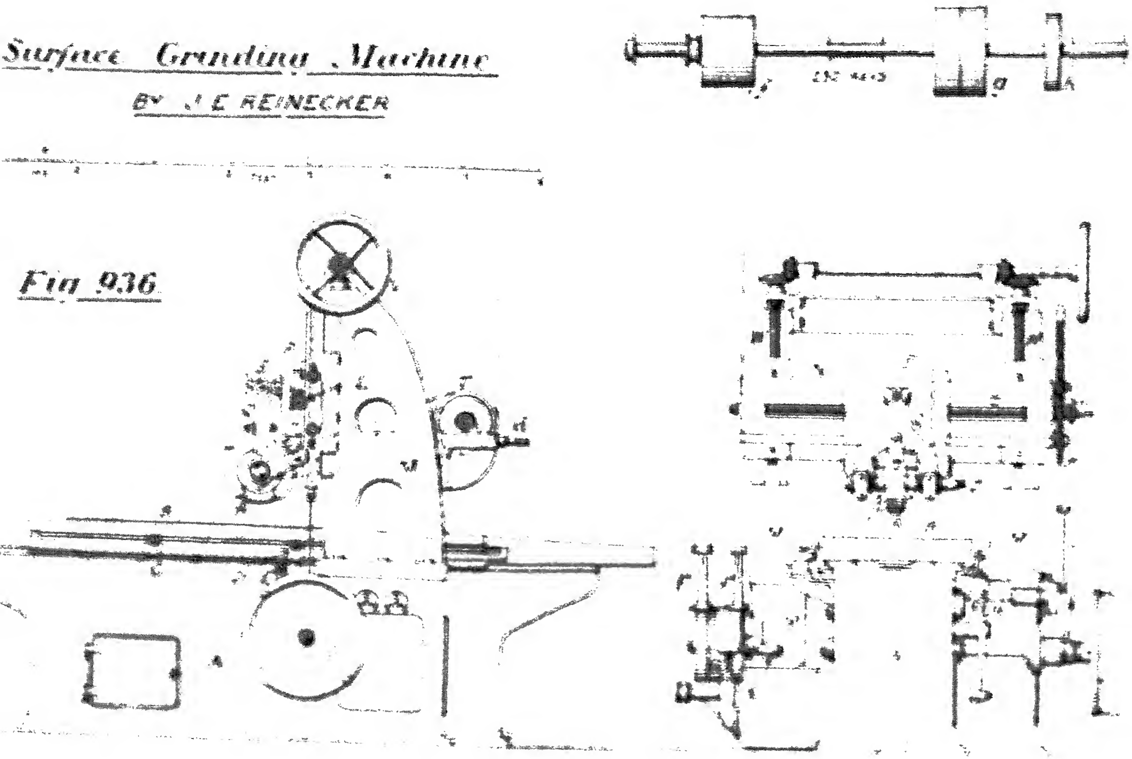
As the work to be ground must have been roughed out in a lathe, a planer, a slotting machine, a boring machine, or some other well-known machine tool, it is evident that the general form of the grinding machine for each purpose must partake of the character of the particular roughing machine adopted, and thus we have cylindrical, surfacing, vertical-spindle, and internal grinders. All the tools to be described are the manufacture of Mr. J. E. Reinecker, and the drawings have been kindly supplied by Messrs. Pfeil & Co.

The Cylindrical Grinding Machine, shewn in Fig. 935, is naturally of lathe form. Upon a bed A is placed a long table B, for the purpose of carrying the driving headstock C and the poppet head D; and the work is rotated between the centres by the pulley E through the carrier F, the drive being from the first countershaft K through cone pulleys J H and long drum G. The slide or table B moves to right or left for a somewhat rapid traverse, and carries a second table L, to which the headstocks are bolted. This upper table may be swivelled through 15° on either side of the long axis, round a central stud, by means of the tangent screw J, for the purpose of accommodating tapered work, but is fixed to the table B by end bolts when grinding is proceeding. At each end of the traverse stroke the tappets M M alternately engage the lever N, which puts one or other of the bevel wheels P P in gear and so reverses the table, the positive motion of the clutch being secured by the action of the spring stop *k*; and the drive is from countershaft K through cones Q R and thence by bevels P P to a pinion and rack underneath the table. At each reversal the lever N also strikes a second lever S, which turns the ratchet wheel T through spur gear, thereby causing the emery-wheel carriage to be drawn nearer to the work,

Surface Grinding Machine

BY J. E. REINECKER

Fig. 936.



increasing the depth of the cut; and it will be noticed that this advance occurs after every stroke, in whichever direction, being adjusted from large to very small by the screw U, or put out of action entirely. The traverse can also be stopped at will by disengaging the stop *h* on the lever N, and the cut advanced by hand. The headstock C can be swivelled on a central pivot and bolted in any new position up to 90° from that shewn, thus permitting the grinding of every variety of chucked work, cylindrical, conical, or surfaced. The emery wheel X is supported in a headstock having two bearings that can be adjusted for wear by the milled nuts at *b*, for above all things these wheels must run truly and without shake. The head is carried on a series of slides, V *p q*, supported on the bracket W, the actual sliding taking place between *p* and *q*; while V can be rotated regarding *p*, and *q* regarding W. As indicated in the figure, the cut advance is across the bed, but if the grinding head be turned round a vertical axis, not only can any new position be maintained up to 90° from that shewn, but the advance can be obtained in any given direction depending on the relation of *q* to W. These changes are desirable when coned work is to be ground as a part of long bars, which would be outside the capabilities of the movements of C or L. The pump *g*, driven from the pulley *f* on the first countershaft, supplies water continuously to the wheel X through the pipe *t*, which returns to the tank by pipe *r*; and wheel *m* is for hand traverse when setting.

Internal grinding, that is the finishing of holes that have been bored, requires a much higher rate of revolution for the emery wheel, because of the very small diameter of the latter, and a head for this purpose is shewn in Fig. 938. The headstock *k*, for the work, is fastened to the table *l*, as already described, and the work itself is supported in a concentric chuck *j*. The grinding head *a* has two spindles, one of which, *c*, is for external work. The rightward end of spindle *c* is provided with a pulley *e* which drives the internal grinding spindle *g* through pulley *f*. The spindle *c*, being revolved as usual by a vertical belt upon pulley *e*, serves, it will be seen, as an additional countershaft, and causes the spindle *g* to revolve at the desired speed. The grinding head in Fig. 935 is, therefore, adapted for internal grinding by

the addition of an attachment carrying a spindle similar to *g*, while a pulley like *e* is placed on the existing spindle. By this apparatus holes of a depth of 40 ins. have been trued, and provision is made for various diameters to a minimum of $\frac{3}{8}$ in.

The *Surface Grinding Machine* in Fig. 936 is built on the lines of a planing machine. Thus, on a bed A, a table B travels backward and forward, to which is fastened the work to be ground. At each reciprocation of the table the tappets C or D strike a lever E, which, acting by cranks and rods on the forks F F, put in gear alternately a crossed or open strap connecting the pulleys at G with the countershaft pulley *f*, from which the table is driven by rack and pinion in the ordinary manner. By the same tappet action the shaft H is rotated through a spur-gear connection, and a connecting rod J moves a rack K, which in turn moves a ratchet L by a pawl, causing the grinding head to traverse across the face of the work. The wheel *j* is for hand setting. The standards M M carry a slide N that supports the saddle P, and a forked bracket Q has two bearings for the emery wheel R. When the work is being set, a fine adjustment between the wheel R and the table is provided by means of the screw or clamp S, which turns the forked bracket round the pin *e* as a centre. This movement is also used for the advance of cutting depth, which is done after each complete traverse of the wheel across the table; and as the centres U, *e*, and T are practically in one line, the length of driving belts from drum T to the pulleys U U on the wheel spindle is unchanged by the movement of the wheel bracket. The wheel R is set to the work in the first instance by unbolting the cross slide N from the standards, and raising or lowering it by the hand wheel V and screws W W. The fast and loose pullies at *g* take the main driving belt, and the pulley *h* drives the long drum T for the emery wheel. The water pump X is driven from the countershaft, the return water flowing from the table through pipe Y; and an adjustment for the wear of wheel bearings is obtained by the screw Z. The grinding wheel is driven by two open belts to the pullies U U in order to cause an even pressure on the bearings, and the wheel is reversed at every change of table motion by means of a reversing box, rack and wheel, not shewn in the diagram.

A *Vertical-spindle Grinding Machine*, illustrated in Fig. 937, is of slotting machine pattern. Although primarily designed for hole-grinding, it can be used on either internal or external work, and plain surfacing can also be done within limits. The main casting A supports the work table B, which is rotated by the

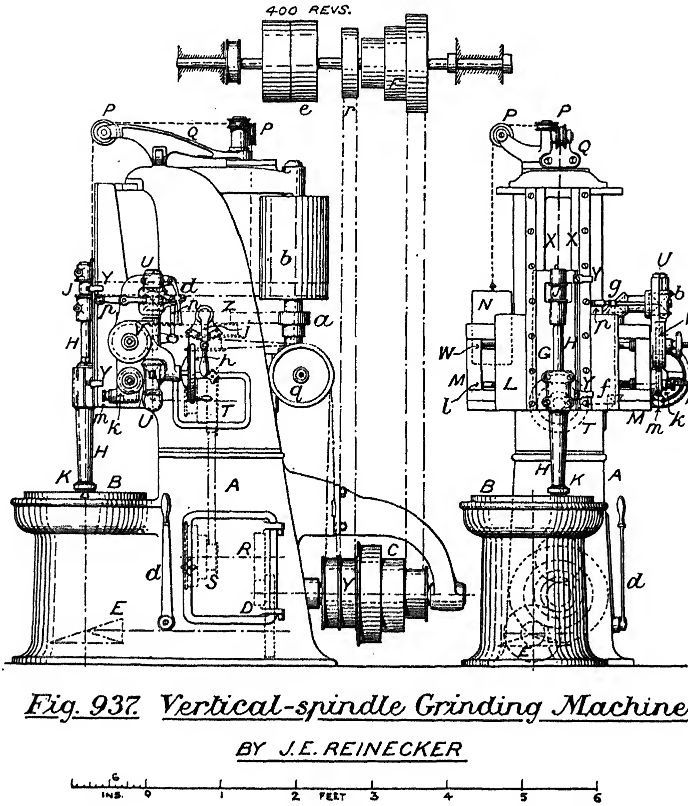


Fig. 937. Vertical-spindle Grinding Machine.

BY J.E. REINECKER

cone pullies F and C through spur gear at D and bevel wheels at E, the variable speed from the cones suiting work of different diameters. The tool head G carries the wheel spindle H H, at the bottom end of which is mounted the emery wheel K. At the top end the pulley J is driven from the drum *b*, whose length allows for the vertical traverse of G between the guides X X. These guides are part of the saddle L, whose position, and that of the

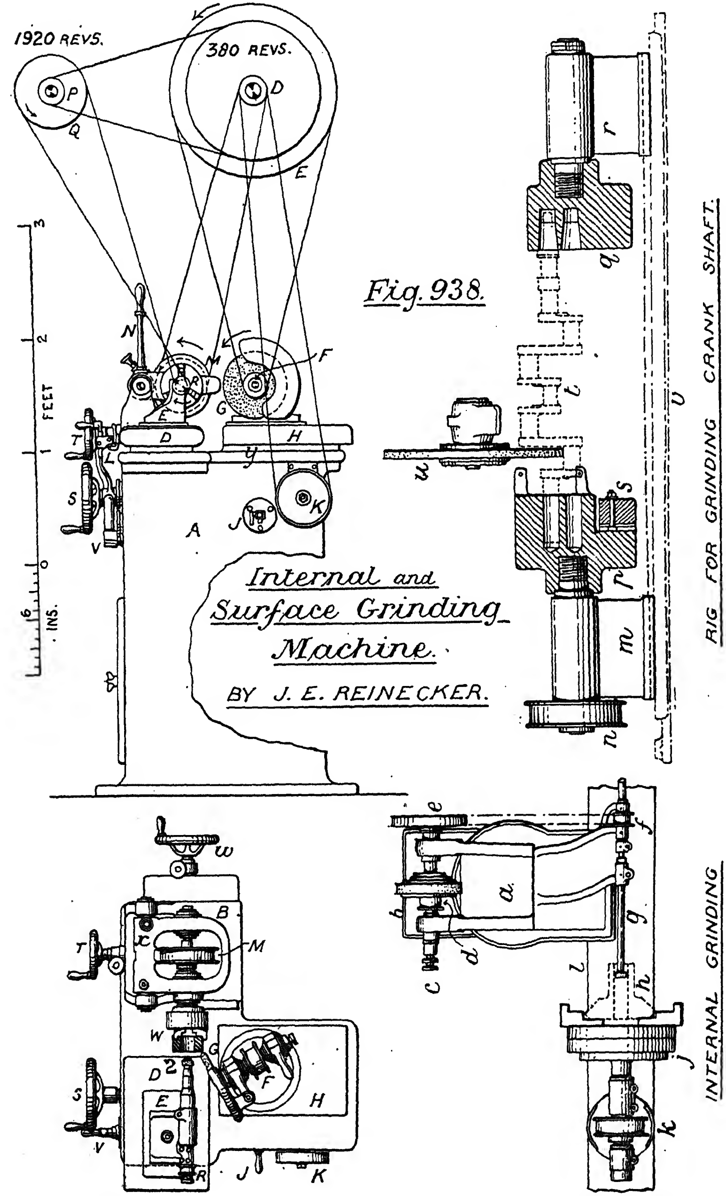


Fig. 938.

Internal and Surface Grinding Machine.

BY J. E. REINECKER.

wheel, can be varied to right or left to suit various sizes of work, and to alter the depth of cut. The wheel head is further balanced by the weight N connected to G by a chain over pullies P P, and the pulley bracket Q can be adjusted horizontally for changed position of the saddle. The emery wheel drive from the countershaft is by pulleys *r* and Y, thence over guide drum *q* to cones Z and *a*, which rotate the drum *b*, and therefore also the wheel spindle: the lever *j* serving to tighten the belt Z *a*, and by unslacking permit a change of speed to the second step of the cone. The spur wheels at R are driven from the main cones, and in turn rotate the cones S and T. This motion is conveyed by the bevel gear *f*, *g*, and *b* to the vertical shaft U U, thence through worm gear at V, and spur pinion on the shaft W, to a rack upon the sliding head G, thus causing the vertical traverse. At each rise or fall of G one of the tappets Y Y strikes the lever *p*, which puts in clutch one or other of the bevel wheels at *b*, thus reversing the traverse, and the quick and certain throw-over is ensured by the action of the spring lever *d*. The worm gear at *k*, being turned by hand through a micrometer wheel *m*, rotates the saddle screw *l*, and thus advances the cut: and the lever *d* puts the bevel wheels E out of gear, and thus stops the table rotation when required. The lever *n* similarly disconnects the bevels *b* to permit of hand traverse by the wheel *h*.

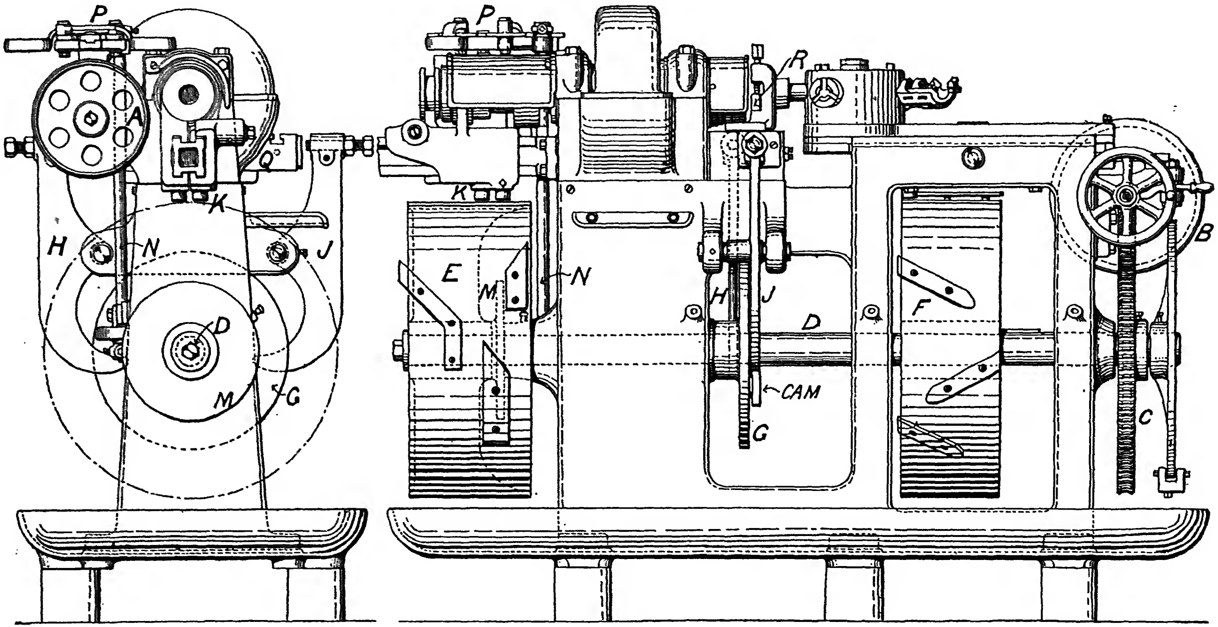
A combined *Internal and Surface Grinding Machine* has been further given in Fig. 938 because of its interest. The machine is not large, but is extremely useful for a number of tool-room jobs. On the bed A are three heads, B carrying the work in a chuck W, while H is the surfacing and D the internal grinding slide. The wheel G is driven by pulley F from the first countershaft D, which also drives the mandrel wheel M; but the small wheel Q and extra countershaft P are provided for the high speed required, the train being E to P, Q to R. The slide H traverses across, and slide D with the mandrel, both motions occurring, backward and forward, simultaneously if required; but the surfacing slide only can be stopped (by lever J) while the machine is in action. The traverse motion is obtained through pullies D and K to three bevel wheels within the bed, as in the previously described machines, and the reverse is effected by tappets on D

engaging with the lever L and putting one or other of the pair of bevels in gear. The head B can be advanced by hand wheels *w* and T, slewed to suit taper holes, or the work may be lifted for examination by means of the lever N.

A grinding rig is also shewn in Fig. 938 for a small three-throw crank shaft. Special heads *m* and *r* are bolted to the table (*see* Fig. 935), and these carry driving bosses *p q*, the weight *s* acting as a balance. The emery wheel must be of large diameter so as to reach the work.

P. 202. **Automatic Turret Lathe.**—Alfred Herbert's Full-automatic Screw Machine is illustrated in Figs. 939, 940, 941, and 942. The first figure indicates by side and end elevation the arrangement of the machine, which obtains its title from the fact that it is largely occupied in the production of bolts, studs, and screws, though various forms of handles and spindles may also be cut from the solid bar or stock, and all this with practically no attention whatsoever. It will be seen from Fig. 939 that there are two sets of driving gear from the overhead countershaft; at A with crossed and open belts so as to rotate by spur gear the hollow mandrel, and at B through worm gear to the cam shaft D. The drum E is supplied with cam plates, which strike the studs at K for the purpose of causing the stock or bar within the hollow mandrel to advance to the correct length, when it is gripped by the collet or chuck; while the cam disc M acts upon a lever, which gives a backward or forward rotation of shaft N, thus moving the forks at P, and producing the forward or reverse rotation of the mandrel, according to which belt is on the fast pulley. Another cam plate G acts upon the levers H and J, that put the slides at Q in position for parting the finished work, the tools being shewn at R, and the turret slide is advanced and withdrawn for the operations of turning, screwing, rounding, &c. by means of cam plates upon the drum F.

Referring now to Fig. 940, which is a section and plan of the headstock: A is the driving pulley, and B C loose pullies, one for the crossed and the other for the open belt, and the fast pulley A is slightly larger than the others, so as to cause a better grip when driving. The lever D, previously mentioned, rocks to right or

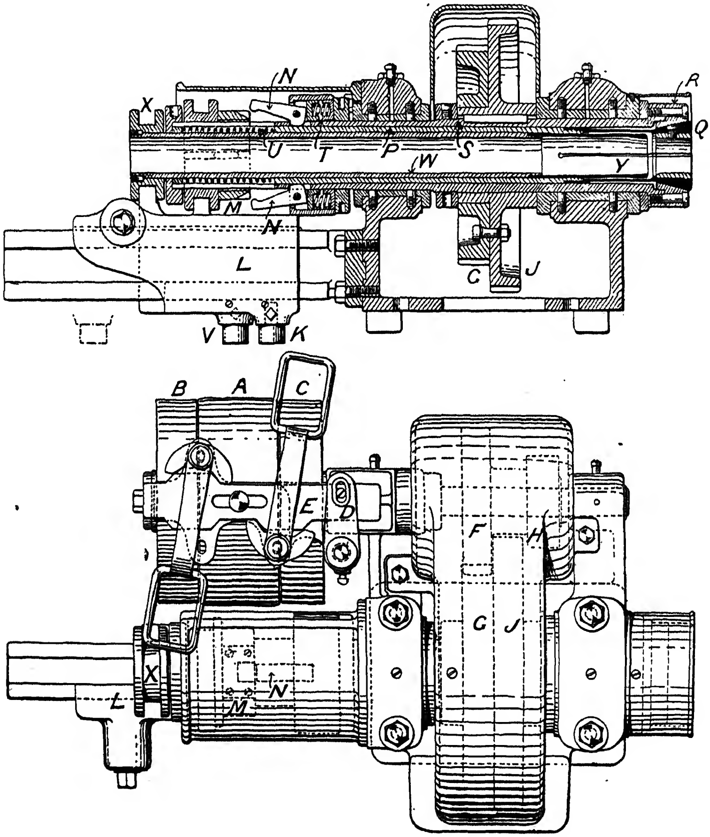


END ELEVATION

SIDE ELEVATION

Herbert's "Full-Automatic" Screw Machine. *Fig. 939.*

left at stated times, thus causing the slide E to move the belt forks and put the one belt or the other in gear as required ,



Herbert's Full-automatic Screw Machine.

The Headstock. *Fig. 940.*

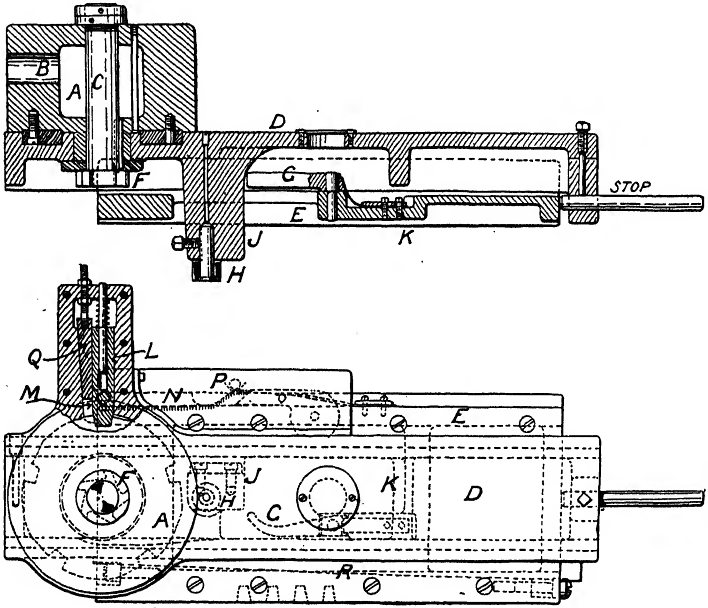
therefore rotating the mandrel in one direction or the other through the spur gear F G or H J, whichever may be in mesh. The stud K (worked by cam drum) is moved rightward to

separate the levers N N and compel them to bite the collet tube P, thus pulling the split cone Q against the hollow cone R on the mandrel tube S, and thereby fastening or chucking the stock bar. A leftward motion of M releases the levers N N, which are then compelled by springs to approach and lie on the tube P; and the latter, being freed, is acted on by the spring U, relieving the grip at Q. The stud V similarly actuates the collar X, and gives movement to the stock tube W, the split collar Y nicely gripping the stock. When W moves leftward the stock is in chuck, and a new position is taken up between W and the stock; but the rightward motion of W takes place only when the chuck Q is freed, and the stock is therefore carried forward for a new series of operations. It will be noticed then that one complete article is machined at each revolution of the cam shaft D in Fig. 939.

Fig. 941 shews an elevation and sectional plan of the camshaft driving gear. The pulley A turns the worm shaft B either directly or through epicyclic gear, so that quick or slow rotation of the cams is provided at will; the slow motion being needed when cutting, and the fast when no actual work is being done, thus saving time in operation. When driving slowly the pinions C C are carried round the fixed wheel D and the free wheel E, the latter having one or two teeth more than D; and if clutch F be engaged the worm shaft receives this slow rotation. By releasing F and putting clutch G in connection, the spur wheels turn solidly and cause B to rotate at the same rate as pulley A. To effect these changes automatically a large disc H has a series of lugs, such as J, fixed upon its rim in agreed positions, and these at the proper times strike a pin K on the bell crank lever L, thus compelling the rack lever M to move the shaft B to right or left, and engage clutches F or G alternately, a quick and positive action being secured by the spring stud N. These movements may be prevented when required by releasing the catch P, and the shaft B can then be turned by hand wheel Q through clutch R, which is ordinarily kept out of action by the spring S.

The turret head is given in Fig. 942 by a sectional elevation and plan. The turret A is mounted so as to rotate on stud C when required, and carries in five holes as at B the necessary turning tools for the operations to be performed. The turret

slide D is moved rightward and leftward on the bed E by the roller H from the action of the cam drum. On the bottom end of stud C is a star wheel F, and when the slide D moves to the right the star F catches the finger or pawl G, causing the turret to rotate one-fifth of a revolution, the finger springing out of gear on



Herbert's Full-automatic Screw Machine.
The Turret Slide. Fig. 942.

the return stroke. The back travel is limited by the boss J, which strikes the part K of the bed casting. The angular position of the turret is decided and retained by the spring stop L, which engages with a plate on the base of the turret, and the stop is released just before rotation of turret by the square stud M mounting a cam slide N into the position P. The wedges Q and R are adjusted so as to give steadiness without undue grip.

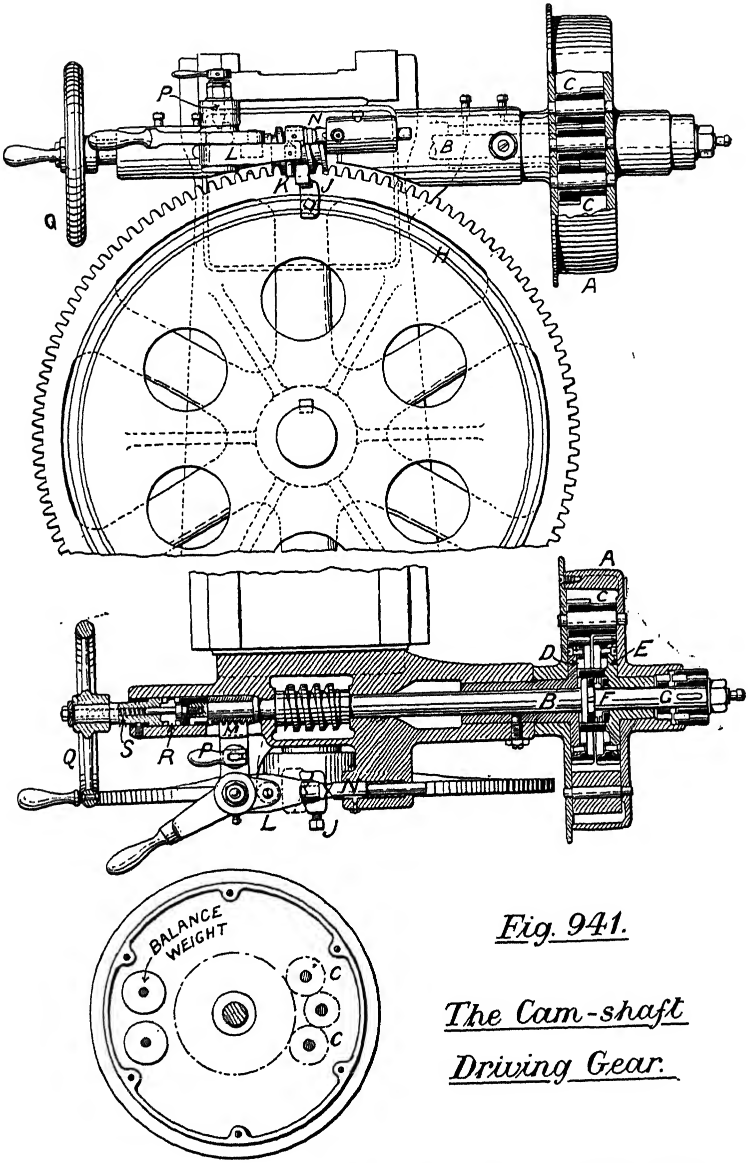
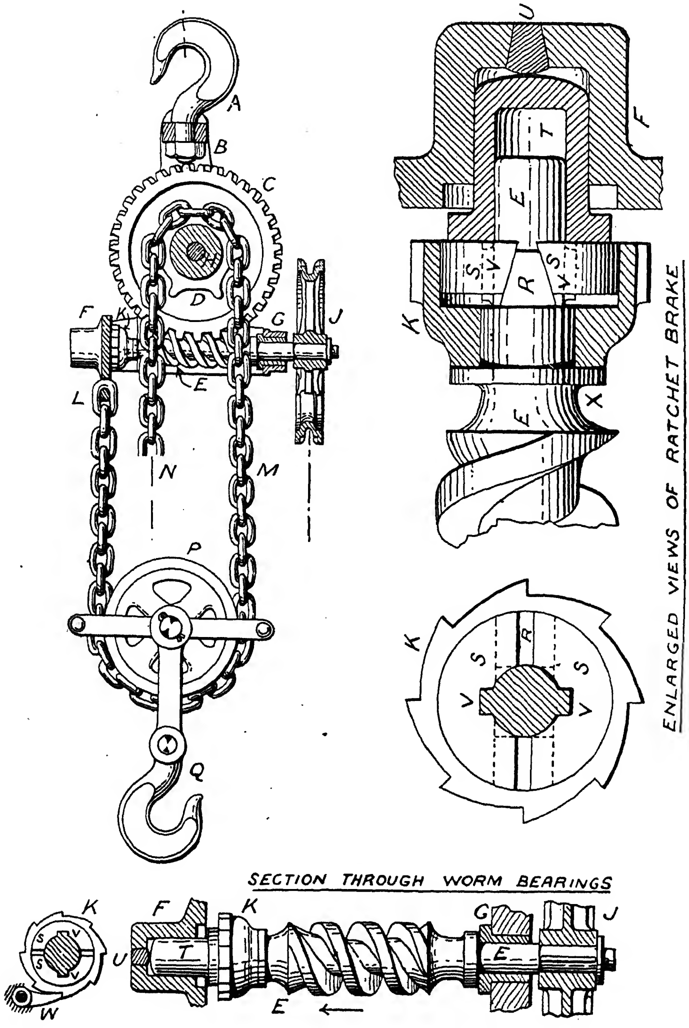


Fig. 941.

The Cam-shaft Driving Gear.

Herbert's Full-automatic Screw Machine.



Worm Gear Pulley Block.

BY HERBERT MORRIS & BASTERT

Fig. 943.

P. 204. **A Worm Gear Pulley Block.**—It has already been stated that in the differential pulley block the load will not overhaul, that is, will remain hanging when the hands are taken off the lifting chain. Let us examine the price to be paid for this in frictional loss. In any machine, where P and P_1 are the theoretical and practical efforts respectively, W the load, and R the velocity ratio,

$$W \times 1 = P \times R \quad . \quad \text{and} \quad P = \frac{W}{R}$$

And the frictional effort $= P_1 - P = P_1 - \frac{W}{R}$

If now we require that the load shall be just balanced by the frictional resistance, we have

$$W \times 1 = \left(P_1 - \frac{W}{R}\right) R = P_1 R - W$$

$$\text{And} \quad P_1 = \frac{W + W}{R}$$

But in that case

$$\eta = \frac{P}{P_1} = \frac{W}{R} \div \frac{2\,W}{R} = \frac{W}{2\,W} = \frac{1}{2}$$

So that, if the load is not to overcome the frictional resistance, the efficiency of the machine cannot be greater than 50%. Indeed, in the Weston block it is only 30%, or 70% of the manual energy is lost in useless resistance. The Moore block, p. 525, is somewhat better, having an efficiency of about 40 to 45%. A well-cut worm gear will give yet more economical results, but is not proof against overhaul at any velocity ratio equal with that of differential blocks. The difficulty is overcome in Herbert Morris and Bastert's pulley block, Fig. 943, by an automatic friction brake wheel, which sustains the load when standing, but does not absorb any portion of the energy of lifting. The supporting hook A is attached to a bracket B in which are bearings H, F, and G for the worm wheel C and the worm E. The sling hook Q and snatch block P are supported by a chain N M, whose standing end L is connected to the bracket B, while the free end N is of any convenient length for the height of lift required; and this chain is raised by means of the sprocket D fixed to the worm

wheel. An endless pulling chain lies in the sprocket J, which is keyed to the worm spindle; and the velocity ratio in the example shewn is about 34 : 1; for there are 34 teeth in the worm wheel and two threads on the worm, making 17 : 1, and the radii of J and D are as 2 : 1; so the total ratio is $17 \times 2 = 34 : 1$. Taking an efficiency of 75%, the real mechanical advantage is $34 \times {\cdot}75 = 25{\cdot}5$, and a pull of 65 lbs. by one man will lift $25{\cdot}5 \times 65 = 1660$ lbs. or nearly $\frac{3}{4}$ of a ton. Referring to the large section in Fig. 942, the thrust of the worm is received in two places, firstly between the collar X on the worm spindle and the steel bush T, and secondly between the bush T and the hard steel pivot U. The former causes the ratchet socket K to bear against two wedges at R, which separate the gun-metal segments S S, and compel them to bind on the interior of K with a grip proportionate to the weight lifted. As the segments are held to the worm spindle by keys V V on a square on the spindle, the overhaul on the worm is evidently prevented by this frictional grip and the resistance of the pawl W, shewn in lower view; but if the workman desires to lower the weight he can do so by pulling on the lowering side of the chain on J, when the segments S S will slip, the work of overcoming their friction being proportional to the load itself. Thus a light load can be lowered several feet by one vigorous pull on the hand chain, but a heavier load is proportionately resistant. Now, although the frictional resistance is always present, and immediately ready to exert itself towards safety, it is entirely removed on the raising of the load: for the parts E, S, K, and T then revolve as one, the ratchet wheel clicking freely over the pawl; and the only resistance is then the friction of the pivot U, whose diameter is seen to be small.

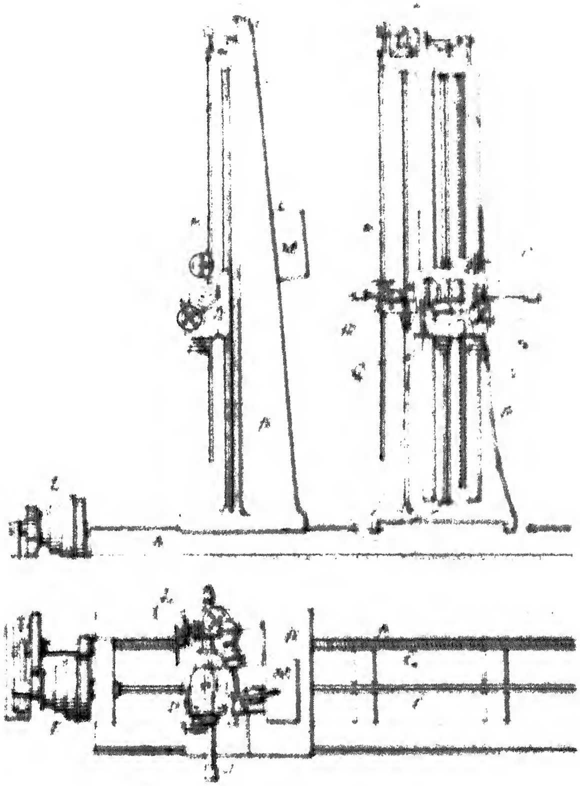


Fig. 944

Drilling Machine for Marine Boilers

By Messrs. Geo. Booth & Co.

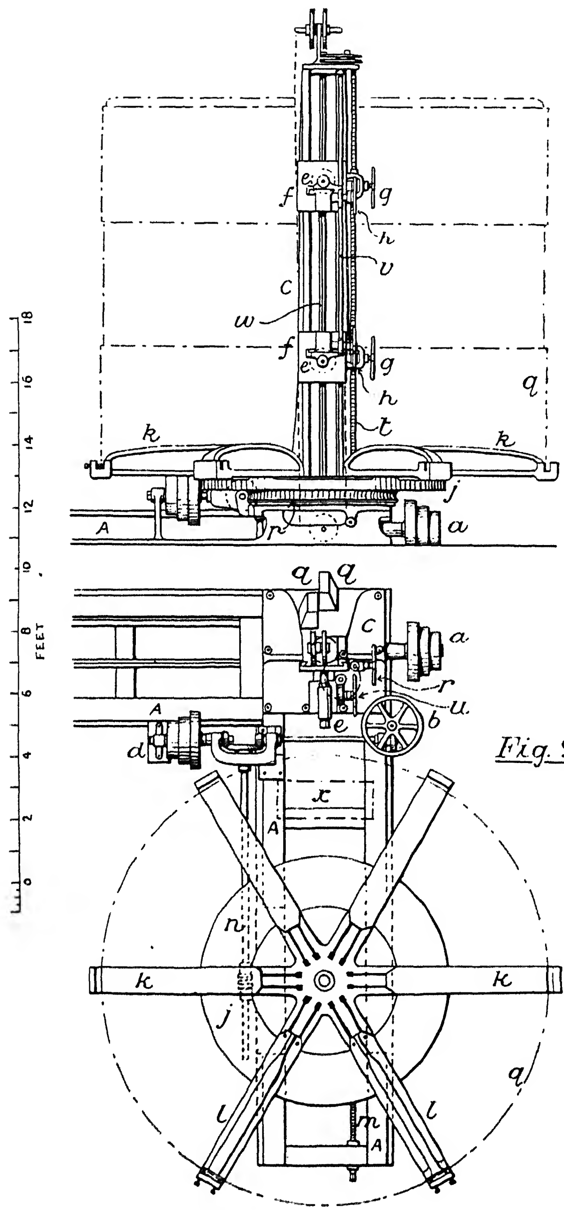


Fig. 945.

CHAPTER VII.

Pp. 309–10. **Drilling Machine for Marine Boilers.**—The methods of drilling large marine boilers 'in position' was fully described at these pages, and illustrated by Plate XIII., where the boiler, with its axis placed horizontally, was laid in a suitable cradle. In Figs. 944-5 are given some views of a drilling machine built for two purposes: first, for drilling boiler shells when the boiler axis is vertical, a method having certain advantages over that of Plate XIII.; and second, for drilling the flat ends of a boiler whose axis is horizontal. The bed of the machine, A A, is of L-shape in plan, and there are two vertical standards to carry the drills, one of which, B, must be traversable along the bed to a range equal to the largest boiler diameter. The other, C, is fixed, and horizontal adjustment deputed to the boiler table *j*. The advantage of one bed for the two standards lies in economy of driving power and attendance. The standard B is traversed for new position by the screw G, operated either by hand or power, the former by the cross-handle L and the latter from the cone E. The drill saddle D, carrying the drill spindle J for drilling or tapping, is balanced by the weight M, and can be raised or lowered by hand through the wheel N, or by power through the shaft K, reversal being obtained by clutching opposite bevels at T. The drill spindle is driven from cone E and shaft F, being reversed by the hand wheel P, which turns a worm-segment lever that puts one or other of the bevels Q in gear, and the spindle is adjusted for depth by the hand wheel R. The feed cones are seen at S.

The fixed standard C carries two saddles *f f* for the drill spindles *e e*. They are driven by another countershaft through the cone *a* and vertical shaft *w*, and the saddles are balanced by the weights *q q*. The drill saddles are raised or lowered by the screw *t*, which is moved by the cross-handle *r*; and the feed is obtained either automatically from the shaft *v* and worm gear to each spindle, or can be worked by hand through the cross-handle *u*. The boiler shell is supported on six arms *k k*, which are bolted to the table *j* in positions that suit the boiler diameter, and the final clamping is done by set screws on the two arms *l l*.

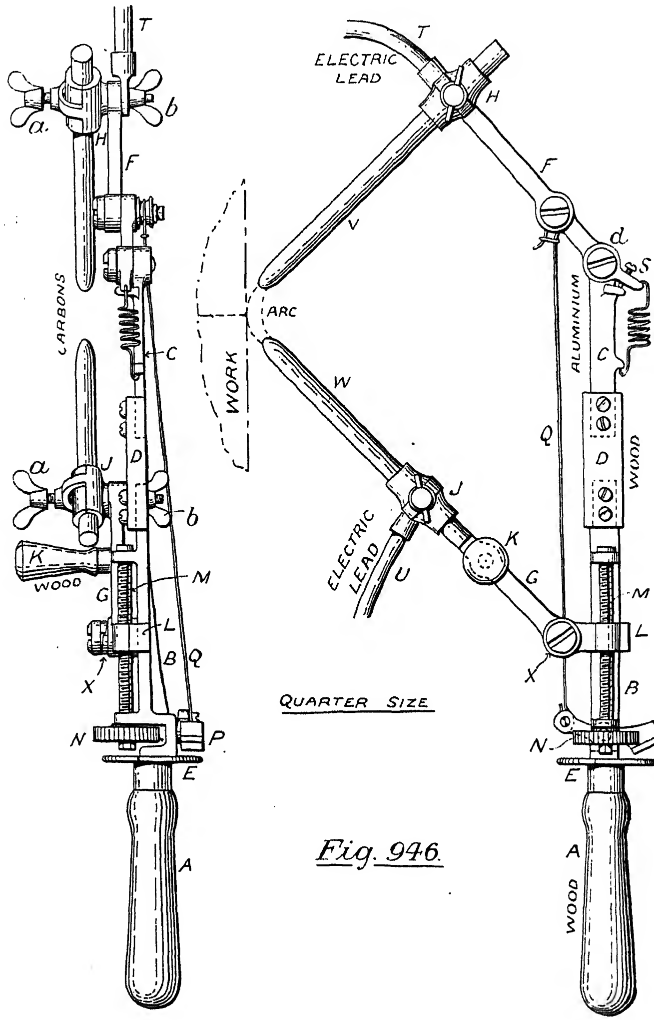


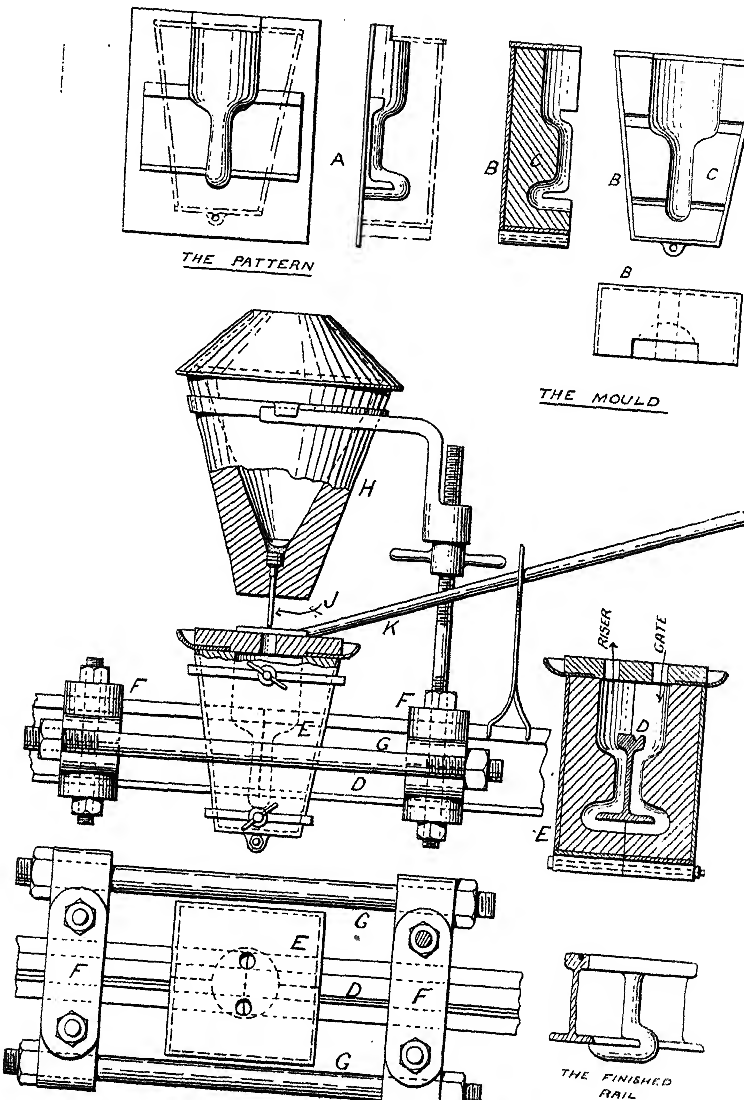
Fig. 946.

The "Voltex"
Electric-Welding Apparatus.

The boiler is rotated to new positions by the hand wheel *b*, which turns a worm shaft *n*, while screw *m* moves the whole boiler nearer or further from the drill. This machine is also used for turning the boiler ends. For this purpose a compound tool-rest is fixed in the position *x*, and the boiler is then rotated continuously by power, from the cone *d*.

Pp. 330 & 827. **The Voltex Electric-welding System** has already been briefly explained at p. 827. It remains to describe the apparatus itself as illustrated in Fig. 946. A wooden handle A carries the main frame B D C, the parts B and C being of aluminium, and D an insulator of wood. The two aluminium levers F and G hold the carbons V and W in clamps H and J, the thumb screws *a a* serving to fasten the carbons, and *b b* the electric leads T U from the dynamos. The lever F is pivoted at *d*, but is held in the position shewn by the spring R and stop S, the latter being slightly adjustable. The arm G can be set at any convenient angle by the handle K, a fine adjustment being obtained, even when current is passing, by the vulcanite wheel N, which is turned by the thumb, and thus moves the nut L along the screw M. When set, the arm is held rigidly by the frictional grip of a spring washer X. The carbons usually form an angle of 45°. Being approximately set to the proper distance apart, the arc is struck by pressing the lever P with the thumb. This lever is pivoted on B, and is faced with vulcanite, and the hand is further protected by the vulcanite shield E. When P is pressed, a pull is caused on the wire Q, and the carbon L is thereby brought in contact with W. On release, the spring R separates the carbons, and the arc is formed, the best length of which is found by trial, adjusting at N. The workman's eyes are always protected by goggles of blue glass.

P. 330. **Thermit Welding.**—It had long been known that aluminium was useful in reducing metallic oxides, entering into combination with the oxygen at a very high temperature, and leaving the metal free; but it remained for Dr. Hans Goldschmidt to utilise the great heat of combination as a means of welding iron or steel, and thus provide a formidable opponent



Thermit Welding.

Fig. 947.

of electric-welding processes. Introduced into England in 1902, thermit has been largely adopted for the welding together of the ends of the rails of street tramways. Welded rails have been always greatly desired, but the great difficulty was the question of expansion and contraction due to the daily changes of temperature, which forbade the adoption of continuous rails on the ordinary railway tracks. Rails in tunnels, however, and those of street trams embedded in the non-conducting earth, are not subject to such extreme temperature variations, and a forcible holding down during those changes will only introduce a longitudinal stress of between one and two tons per square inch, a matter of no real consequence. The advantages of welded rails are, on the other hand, great, both as regards smoothness of working and electrical continuity. The welding was formerly performed electrically, but it may be confidently asserted that thermit welding has entirely superseded the older method, both on account of its simplicity, and the great cheapness of first and working costs.

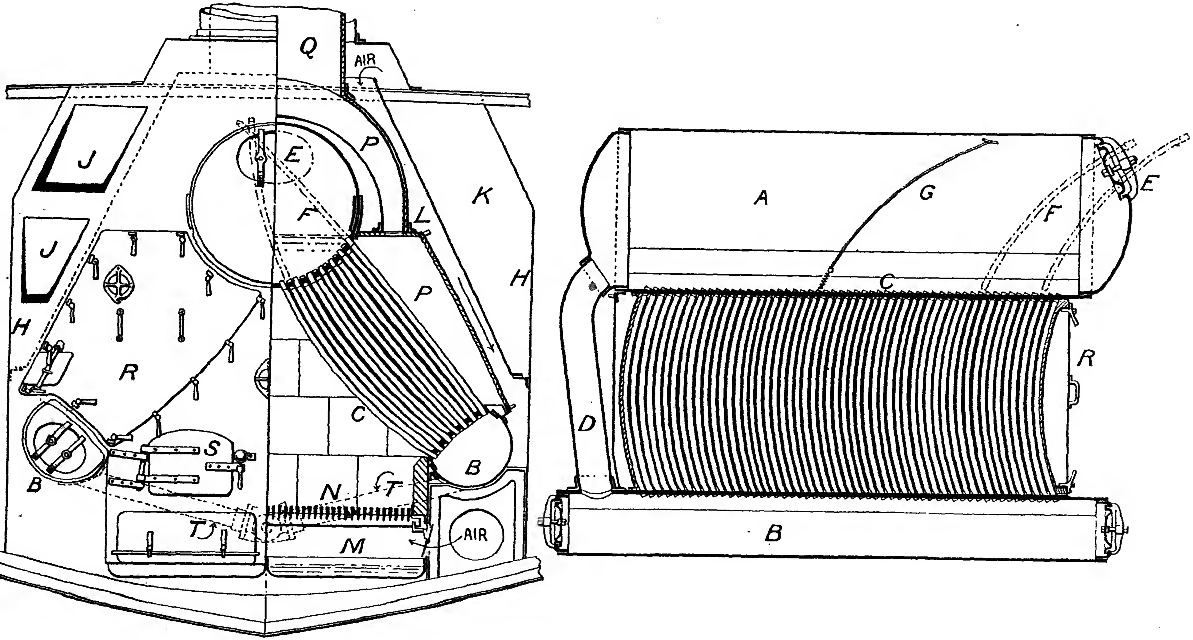
Thermit is a granular mixture of aluminium and iron oxide, in exact chemical proportions between the aluminium and the oxygen; so that a perfect union may ensue on combustion, and nothing but alumina and pure iron remain behind as products. Its temperature of ignition is a little less than that of molten steel, and below this it will not ignite. It is perfectly safe, therefore, against a red-hot poker, or even molten cast iron; and may be thrown into an ordinary fire without injury. It would be ignited, however, by molten steel, or, indeed, by any other method which would produce the temperature of ignition, however small the area of attack. In practice a special ignition powder is provided, which may be started by a lucifer match, thus causing a local heat of sufficient intensity to ignite the thermit, whose ingredients now combine and cause the enormous temperature of 5400° F. by their combustion. The iron being freed, the aluminium goes to form with the oxygen a slag of alumina, which appears as thin, dark red flakes of what may be called emery, ruby, or sapphire, both slag and iron remaining fluid at the high temperature. It is this immense evolution of heat that serves for the welding of two pieces of iron or steel, being thus analogous to electric welding,

or, indeed, to any other process where a sudden supply of intense heat is desired.

We may now explain the actual apparatus adopted in the welding of tram rails; and, firstly, it must be understood that not only are the rails welded, but a portion of iron (from the thermit) is run all round the joint, for which a mould is to be provided. The pattern for the half-mould is shewn at A, Fig. 947, consisting of a copy of the half-rail, the scum head, and the wrapping of molten iron. A box B is fitted over this, as seen dotted, and a mixture of sand and clay is rammed between the box and the pattern to the appearance shewn at C. Two of these moulds having been made, and dried in a portable oven at 600° F. for two hours, the rail D is painted on the tread and sides of the head so as to prevent adherence of metal at these places; and the moulds with their cases are placed one on each side of the rail, and clamped together as at E. The rails having been previously secured end to end by the clamps F F and the connecting bolts G, the mould and rail are now well luted with sand and clay round all the meeting edges, especially at the head. The crucible H, lined with magnesia, is supported with its mouth about 3 or 4 inches above the mould, and the opening is closed by two thin asbestos washers and an iron disc, held up by a $\frac{1}{4}$ in. iron rod J and the spade K. The washers are covered with magnesia to prevent accidental tapping. The crucible is next filled with the thermit mixture, poured from a bag, and a salt-spoonful of ignition powder is placed upon the top. Protecting his eyes with blue spectacles, the workman starts the ignition by means of a lucifer match, covering the crucible, and the combustion is completed in about 10 seconds. He then deftly withdraws the spade after lifting the tapping rod, and the molten mixture runs into the rail mould, firstly the iron, on account of its greater weight, and afterwards the slag, which has equal weight with the iron but three times its volume. Finally, after 3 minutes' wait for the uniform welding heat, the bolts G G are tightened up to cause a true weld between the rail ends; and, after another short rest to cool, the mould and superfluous material are broken away with a sledge-hammer, the whole process having occupied 30 or 40 minutes. The action of the tightening bolts G has caused some

metal to extrude at the rail-head, which must be chipped away when quite cold, and sometimes, for this reason, the drawing up is not performed; but it is certainly advisable, for a true weld of 87% the strength of the solid rail is thereby produced even without the extra band of iron. The latter, however, serves as a most efficient fishplate of a material very similar to mild steel, and is most thoroughly amalgamated with the rail metal. The usual tram rail requires 22 lbs. of thermit (at 10*d.* per lb.) to each weld, containing about 11 lbs. of pure iron. To this is generally added 15% by weight of iron punchings, thus yielding 14 lbs. of iron to the 22 of the mixture, a procedure not only more economical, but also a means of reducing the extremely high temperature within manageable limits. The cost of the process compares favourably with electric welding.

There are three ways of using thermit practically. (1) The iron may be run before the slag, as already described, by tapping the crucible at the bottom, causing the highest possible temperature. (2) The crucible may be tapped from the top, running the slag first, which, meeting the parts to be welded, coats them with a resistant film. This prevents the close contact of the molten iron, and reduces the temperature to a reasonable welding heat between the surfaces of union, the joint being upset by clamps. This method is adopted for iron or steel pipes, and gives a perfectly clean joint after the removal of slag. (3) The rolls of rolling mills often break off at the ends, and are to be patched with an entirely new piece. The broken roll being sunk in the ground, end up, a mould is made from a pattern and placed on the top. The bottom of the mould (or end of roll) is covered to ½ in. depth with molten cast iron, and the whole then filled with thermit at the rate of 30 lbs. to the square foot of joining surface, then ignited in the usual manner. In this way cast iron and steel can be connected, and the surfaces become as one. Section bars of all kinds may be welded by the first or second methods as preferred; while if the third method be adopted for any purpose where molten steel is the material required, the heat of the latter will be sufficient without the necessity of external ignition. Steel examples would be the stern frames of ships, which have been welded and patched very successfully.

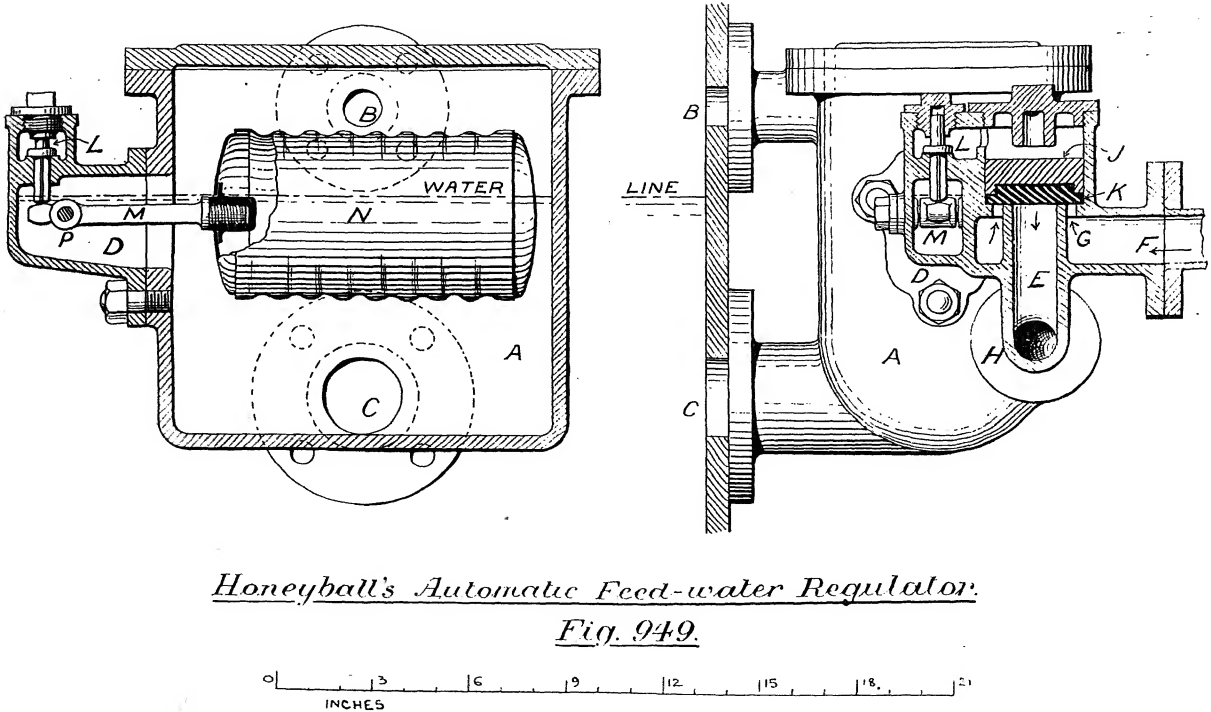


White and Forster's Water-tube Boiler. Fig. 948.

The resulting slag is a useful bye-product, being taken to emery mills and ground into powder; and various important alloys required in steel manufacture are produced by the heat of thermit, as—

Chromium ingot ...	98 to 99% pure	:	free from	carbon
Manganese „ ...	98 to 99% „	:	„	„
Ferro-Titanium ...	20 to 25 T.	:	„	„
Ferro-Boron	25% B	:	„	„
Chromium-Manganese	30 Cr : 70 Mn	:	„	„
Manganese - Titanium	35% T	:	„	„
Manganese-Copper ...	30% Mn	:	„	iron
Manganese-Zinc ...	20% Mn	:	„	lead
Manganese-Tin ...	50% Mn	:	„	„
Nickel-Thermit ...				

Pp. 339 and 832. **The Water-tube Boiler.**—Dealing with the marine type of 'forcing' or quick-steaming boiler, where two mud drums are connected by sets of tubes to a steam receiver so as to form the letter A : the Yarrow and Thornycroft boilers have already been mentioned as examples. The former is provided with perfectly straight tubes, and there is no downcomer, some of the tubes themselves being intended to serve that purpose. The Thornycroft boiler has tubes curved to meet the draft normally, with the object of increasing the circulating efficiency, and curved tubes have been adopted by many other makers for similar reasons, not necessarily normal to draft, but always, it is believed, having a good effect on the circulation. The objections to these are the difficulty of internal cleaning, and the fact that several different forms often occur in one boiler. The White and Forster boiler, Fig. 948, is designed to overcome the objections to curved tubes while retaining their advantages. Referring to the figure, the steam drum A is connected to the mud doors B B by the tubes C C, which are all curved to the same radius, and may be all easily withdrawn through the manhole E as indicated by dotted lines at F. Also the interior of the tubes can be cleaned by one curved brush G. The whole boiler is placed within the outer casing H H, and the air for the fire enters by the flap doors or dampers J J, thence down the passage L, being



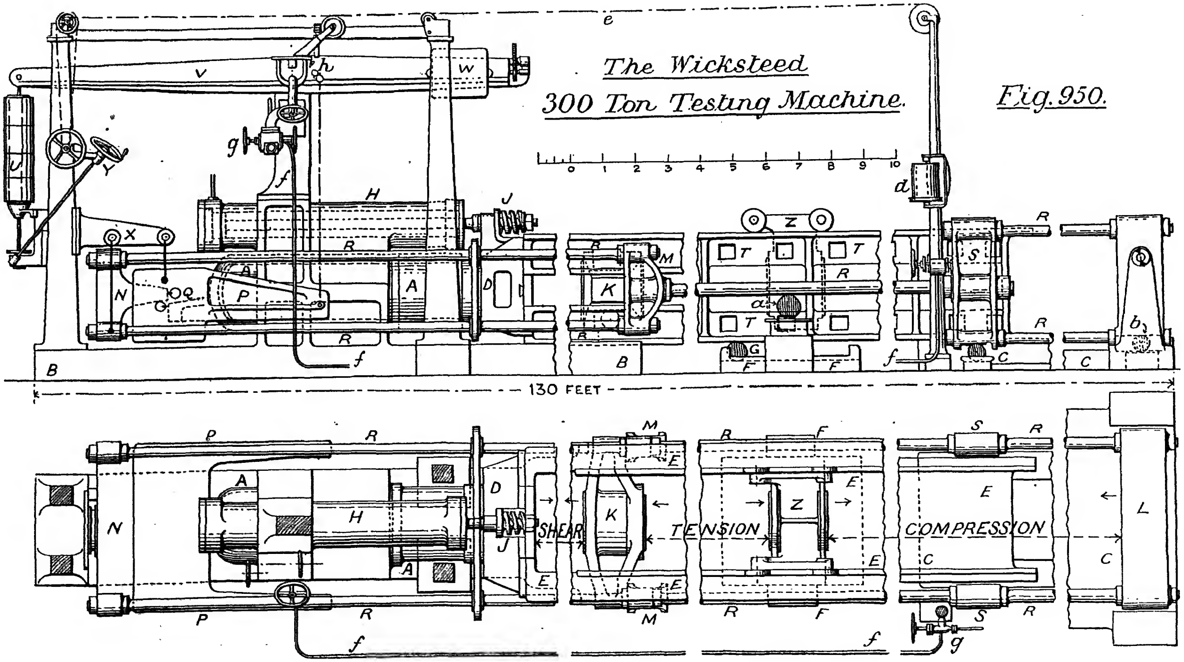
Honeyball's Automatic Feed-water Regulator.

Fig. 949.

heated on its way by contact with the uptake, and in at the ash-grate space M to the fire N. The gaseous products then proceed from N across the tubes C C to the uptake P P and funnel Q. Soot doors are provided at R R, fire doors at S S, and an equalising pipe T T to a blow-off cock in the centre.

Pp. 831 and 918. **Automatic Feed-water Regulator.** —A further example of this interesting contrivance is illustrated in Fig. 949. Mr. Honeyball's invention has the merit of simplicity, though involving the use of a pilot valve as in Mr. Forster's regulator on p. 919. It was originally designed for evaporators, or the small boilers which provide the make-up steam in marine practice, but the principle can be equally well applied to full-sized boilers. A vessel A is connected to the boiler by the tube B in the steam space, and by C in the water space. Attached to the vessel is a gun-metal clackbox or feed check D E, and within this box are two valves, J K being a main piston valve having a 'woodite' face K, and L the pilot valve. The latter is governed by the float N, and the lever M pivoted at P. The feed water enters at F, and, passing through the orifices at G, turns down by the pipe E to the boiler, the incoming pressure lifting the piston J K, a loose fit in its chamber, for the purpose. This kind of action goes on steadily until the float N has risen high enough to free the valve L, which therefore closes. A curious result follows the closing of L, for, whereas the piston J K has up till now been lifted freely by the pressure, the rigid block of water retained in the space from L to J positively resists any attempt to move the piston, and not the smallest portion of water is able to pass, the perfection of the temporary joint at K being enhanced by the very slight elasticity of the woodite. If, however, the float falls, and opens the valve L ever so little, the feed action recommences, and thus a delicate play occurs between closed and open that keeps a steady water level in the boiler.

The diagram shews the closed condition, the supply being automatically stopped till the water level has fallen.



The Wicksteed
300 Ton Testing Machine.
Fig. 950.
130 FEET
SHEAR
TENSION
COMPRESSION

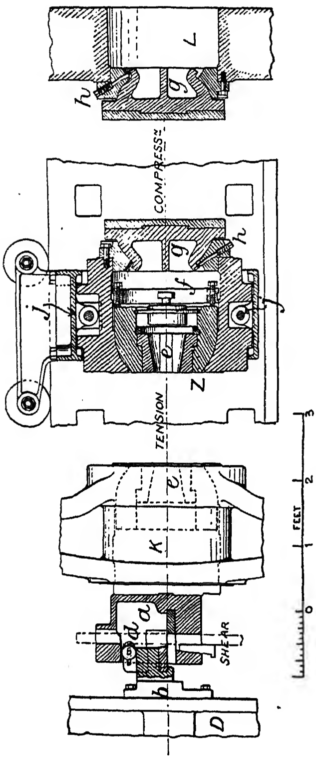
CHAPTER VIII.

Pp. 366 and 385. **Work of Deformation up to Breaking.**—This may be very closely estimated without the necessity of drawing out the stress-strain diagram, using an approximate formula devised by Kennedy. Imagine the diagram divided by a horizontal line passing through the elastic limit, and let r represent the ratio of limit to maximum load, while W is the maximum and total load. Then the lower portion is nearly a rectangle of area $r\text{W}\Delta$. The upper portion may be taken as a parabola whose area is $\frac{2}{3}\left(\text{W}-r\text{W}\right)\Delta$

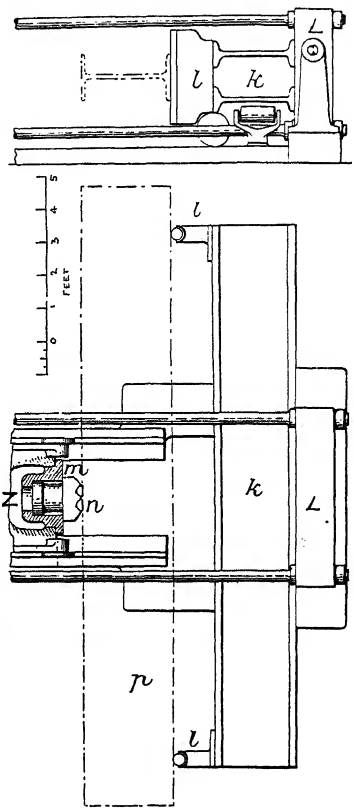
$$\therefore \text{ Total work} = r\text{W}\,\Delta + \frac{2}{3}\left(\text{W}-r\text{W}\right)\Delta = \text{W}\,\Delta\left(\frac{r+2}{3}\right)$$

P. 369. **Coefficient of Linear Expansion.**—Invar is the name given by its discoverer, M. Guillaume, to an almost non-expansible material, consisting of Steel alloyed with 36% of Nickel. Its coefficient of linear expansion has the value ·000,00087 or 87×10^{-8}. This remarkable property of low expansion makes the metal useful for many pieces of physical apparatus, such as clock pendulums, &c. Neither is the expansion anything so important as the coefficient would indicate, for the full alteration in length, as indicated by this value, does not accrue in less than several days, and the real expansion for change of temperature is therefore very small indeed, and practically negligible.

P. 372. **Testing Machines.**—A very fine example of a Wicksteed machine has been built for the French Government. It is a legitimate descendant of the Wicksteed-Kennedy machine on p. 371, and can be best understood by a previous reading of p. 372. It is illustrated in Fig. 950, and has 300 tons load capacity, while admitting specimens 88 ft. long and 3 ft. 3 ins. diameter. In fact, it may be at once explained that the machine is not merely to demonstrate, or to test small specimens representing the material of structures, but is for considerable portions of the structures themselves, such as built-up pillars or girders, riveted joints, &c. It is hoped that by its aid much information



THE STRAINING HEADS
ARRANGED FOR
DIRECT LOADS.



ARRANGEMENT FOR
BEAM DEFLECTION.

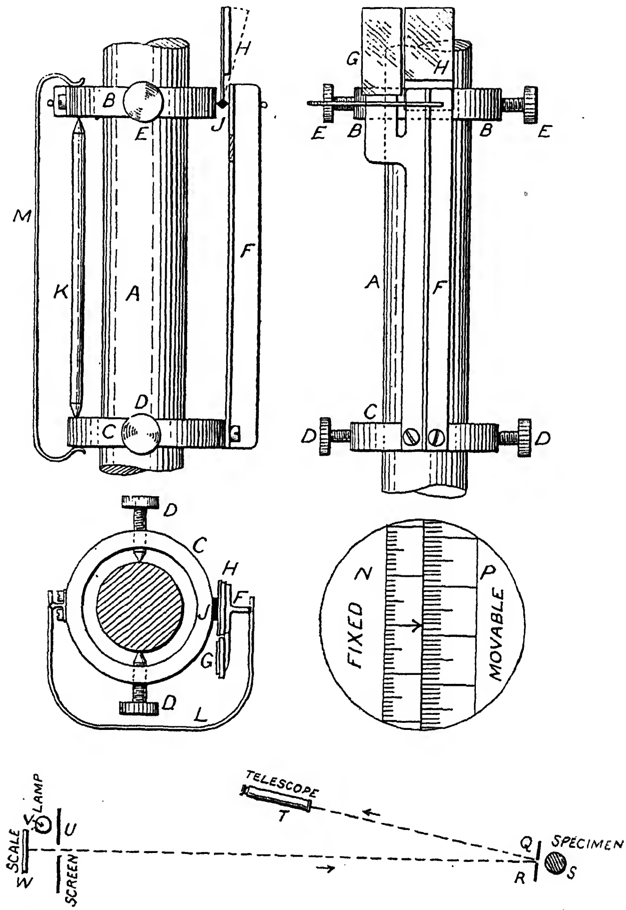
The Wicksteed 300 Ton Testing Machine.

Details : Fig. 951.

may be obtained regarding compound stress in such pieces, and thus be a means of correcting or verifying the theories on which we now rely.

The bed B B supports the straining cylinder A, the plunger of which is connected by the head D to a sliding trough E E that carries the straining head Z. This head is run on rollers to any required position, and is then fixed by large square bolts that are shot out by right and left-handed screws into the square holes T T. The weight of the sliding trough is also taken by rollers as at G, of which there are several, resting on rails F F; and the whole, thus far, constitutes the straining system. The weighing apparatus consists of a steelyard V and counterbalance W, connected to a bell-crank lever P, which bears against a crosshead N, the total leverage being 600 : 1; and the weighing load is further transmitted by the tension rods R R and crossheads M S, to the straining heads K and L, all supported on rollers. As the head Z always moves rightward under the hydraulic pressure in cylinder A, and the heads K L tend to move leftward by virtue of the steelyard load, the positions for specimens in shear, tension, or compression will be as indicated in the figure. It should be mentioned, however, that the head L is also used for beam deflections, when a special support for the beam is placed horizontally across the bed C, and the head Z is brought to a convenient position for the experiment. The whole machine is very long, some 130 ft., so it has been found convenient, in the diagram, to break the picture at four places, for the actual figure would otherwise have been three or four times the length of what is here shewn. The trough being very heavy, is moved backward and forward to starting position, by a piston contained in the smaller cylinder H, and connected by a rod to the lug J on the trough. This method economises the high-pressure water, which at 1700 lbs. pressure in cylinder A of 26 ins. diameter would otherwise be a great loss. The lever P and crosshead N are slung from the bracket X; and the fulcra at Q are 2 ins. apart, so are easily verified. The jockey weight W acts as counterpoise at first, but being moved leftward to gradually balance increasing load, represents 60 tons on the specimen when in extreme position. It is then exchanged for one of the weights at U, which is lifted and put in place by the

hand gear at V; and the jockey is run back to zero. The operation is repeated for another 60 tons stress, and so on up to the full load. An automatic diagram is taken on the drum *d*, the



Morrow's Mirror Extensometer.

Fig. 952.

rise of the pencil being proportionate to the travel of the jockey weight, transmitted by water through pipe *f*, and governed by the valves *g g*. The rotation of *d* indicates the deformation, being actuated by the pull of the wire *e*.

Fig. 951 shews the various straining heads as arranged for the tensile, compressive, shear, and deflection tests, the letters D, K, Z, and L corresponding with Fig. 950. For the shear test a plunger *b* is fixed to the main ram D and enters a socket *a* attached to the head K. Each of these carries a knife or shearing surface, and the two are compelled to slide closely by the action of the roller *d*, the specimen being meanwhile wedged in the socket *a*. Tension sockets *e e* are supplied to the heads K and Z, and the hemispheres are adjusted to true axial line by the set screws in the fixed ring *f*. Compression plattens *gg* are similarly placed in Z and L, also adjustable on spherical surfaces by the screws *h h*, and the screws *j j* serve to shoot out the square bolts for fixing the head Z, as before mentioned. For deflection tests the double girder *k* is laid across the bed, the thrust being received on the head L. A special socket *m* is put in the head Z to receive the pressure foot *n*, which is supplied with two half cylinders to distribute the pressure and avoid indenting; and the specimen beam *p* is supported on loose cylinders contained by the brackets *l l*.

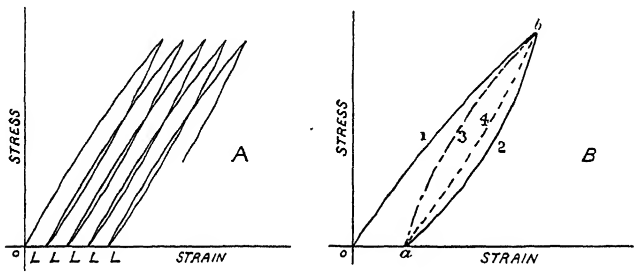
For torsion tests an apparatus is provided which is essentially apart from the machine, consisting of a worm gear not unlike Fig. 336, p. 379.

P. 384. **Strain Measuring.**—All instruments that are devised for the purpose of measuring the minute deformations within the elastic limit of test specimens are termed *Extensometers*, whatever the nature of the strain that is being recorded. They naturally divide themselves into two classes—viz., those where the magnification is performed entirely by mechanical means, and those where the agency of optics is more or less invoked. Under the former are included all forms of lever extensometer, as well as the screw micrometer illustrated by Unwin's apparatus on p. 382, the accuracy of these being generally about one 10,000th of an inch. Under the optical class may be mentioned Ewing's extensometer, where the mechanical magnification is 2 : 1, after which the

readings are taken on a micrometer scale in the eyepiece of a microscope to an estimate of one 50,000th of an inch. In mirror extensometers a beam of light is deflected by a mirror that is tilted by the strained specimen, the deflection being meanwhile measured within a telescope. The apparatus to be described can be read to an accuracy of one 1,200,000th of an inch, which is an immense advance upon that of other forms. Naturally such precision is not required in commercial instruments, but is nevertheless of great value in laboratory operations. Two important requirements must be noticed as belonging to all extensometers of great delicacy: the measurements must be taken on the exact centre line of the specimen, and the instrument must not be touched by hand after being once set. Dr. John Morrow has devised a most successful extensometer embodying these and other principles. Referring to Fig. 952, two loose rings B and C are held on the specimen A by the set screws E E and D D, round which they pivot, and the pointed distance-piece K also provides two other pivots between its ends and the rings. A rigid bracket F is fixed on the lower ring, to carry a fixed mirror G, and a diamond-shaped prism J constitutes a pair of knife edges that support the tilting mirror H. The whole apparatus being bound together by the springs M and L, the experiment is arranged as in the plan view shewn below. A scale W, placed parallel with the axis of specimen S, is illuminated by a lamp V, and a reflected ray travels through the screen U to the extensometer mirrors, strikes them both evenly, and is further reflected to the telescope T. Considering the upper diagram again, the extension of the specimen between E and D causes the rings to pivot round the ends of K, thus inducing relative motion between the ring B and the bracket F, and tilting the mirror into the dotted position. The appearance of the scales in the telescope will then be as shewn at N P, the tilt having caused the P reflection to move downward, while N remains fixed. The scale is divided into 40ths of an inch, and a tenth of each division is easily estimated. Also the scale being set at a distance of 80 inches from the mirror, the magnification without telescope is 3000 : 1; so the smallest reading is $\frac{1}{40} \times \frac{1}{10} \times \frac{1}{3000} = \frac{1}{1,200,000}$ of an inch.

Pp. 385 and 837. **Stress-strain Diagrams:**

Mechanical Hysteresis and Fatigue.—It has been suggested that the phenomenon of mechanical hysteresis is an explanation of the fatigue of materials under variation of stress, and of the slow accumulation of permanent set. Under this theory the diagram A, Fig. 953, would indicate what might be expected to occur. From previous statements at p. 837, it will be seen that on the first and each subsequent stressing and unstressing the ascending and descending curves of stress-strain are mutually concave, and when the base line is reached there is a horizontal



Mechanical Hysteresis and Fatigue.

Fig. 953.

difference L between the curves that may be termed lag. The said theory supposes that if a sufficient interval of rest be not allowed for the lag (being most probably a heat effect) to disappear before a new increase of stress is applied, the lags would be cumulative, and after a sufficient though perhaps an exceedingly large number of repetitions the material would reach the plastic stage, and a *permanent* set would ensue, resulting in final rupture.

Experimenting to prove the truth or otherwise of this theory, the Author has found that it is not borne out in practice; but that the diagram B, Fig. 953, shews the actual result. The first stressing gives an elastic line O *b* numbered 1, and the first unstressing traces the curve 2. Allowing no time for rest, the next stressing

is 3, and the subsequent unstressing is marked 4. It will be seen, therefore, that the lag does not increase, and that the final curve for rapid stressing and unstressing without rest becomes a straight line joining *a b*. Evidently hysteresis is not an explanation of fatigue, and the old fear that continued variation of stress without rest must ultimately end in rupture, however strong the material, is certainly not supported by experiment. In fact, the results of all later endurance tests is to shew that a piece can be made strong enough to stand unlimited and continuous changes of stress without breaking.

P. 388. **Notes on Testing.**—We shall here give a brief statement of the directions to be followed in preparing, conducting, and recording a tensile test.

Preliminary:

1. State the kind of material to be tested.
2. The specimen having been prepared, mark one-inch divisions along its length up to 10 ins. and centre-pop these carefully.
3. Measure the cross-sectional area at several places and note the average.

Fixing the Specimen:

4. Place the specimen between grips in the shackles.
5. Set the jockey weight to zero, with lever balanced.
6. Put a slight tension on the specimen in order to tighten the grips.
7. Attach the extensometer, setting to the extreme or 10 ins. marks.

During the Test:

8. Take extensometer readings at every quarter ton of load, and tabulate load, extension, remarks. Keeping well within the elastic limit, these readings may be repeated from zero.
9. Entering the plastic stage the extensometer is removed. Now book loads and extensions at every half ton increase: the extensions being measured by dividers and taken on a well-divided rule. A little time should be

allowed at each observation for the extension to reach its true value, and the lever must be kept floating between the stops.

10. Note the maximum load just before the contraction commences appreciably. If possible note the actual contraction at this stage.
11. When contraction commences in real earnest run the jockey weight back to zero, and then re-advance it gently so as to only just balance the stress.
12. Note the breaking load, which will be generally much less than the maximum. Remove the specimen.

Results:

13. Measure the final dimensions as regards length over extreme marks, and at the two inches round the fracture: also the contracted area.

We are now to state (*a*) Elastic limit as load and stress, (*b*) Maximum load and stress, (*c*) Extension per cent. in 10 ins. and in the 2 ins. at fracture, (*d*) Reduction of area per cent., (*e*) Modulus of elasticity. The first step is to carefully plot the stress-strain diagram in two parts, one shewing the whole life of the bar, and the other shewing the elastic stage up to yield point with a large scale of extension.

(*a*) The **Elastic limit** may be discovered from the second diagram, being a little below the yield point, and indicated by a slight curvature from the straight line. Dividing the elastic load by the original area of the bar will give the elastic stress per square inch.

(*b*) The **Maximum load** is found from the first diagram as well as from the figure obtained during experiment, and the maximum unit stress may be expressed both in terms of the original area as well as of the area as contracted on the measurement of this load.

(*c*) The **Extension per cent.** is found both on 10 ins. and on the 2 ins. of fracture, as

$$\frac{\text{length after fracture} - \text{original length}}{\text{original length}} \times 100$$

(*d*) **Reduction of area per cent.** will be obtained from

$$\frac{\text{original area} - \text{contracted area}}{\text{original area}} \times 100$$

(*e*) The **Modulus of Elasticity** will be most easily obtained from a large scale plotting of the elastic line. If the plotted points are not truly in line, the best straight line must be drawn through them and

$$E = \frac{\text{unit stress}}{\text{unit strain}} = \frac{\text{elastic load}}{\text{area}} \times \frac{\text{original length}}{\text{elastic extension}}$$

and the result may be stated in both tons and pounds to the square inch.

The preceding directions will be a guide also to tests in compression, strain, and deflection. In compression of ductile materials very little information can be obtained, however, beyond the elastic limit, for the specimen merely flattens out, and does not fracture.

Pp. 385 and 839. **Influence of Temperature on Strength.**—The behaviour of steel as its temperature is increased

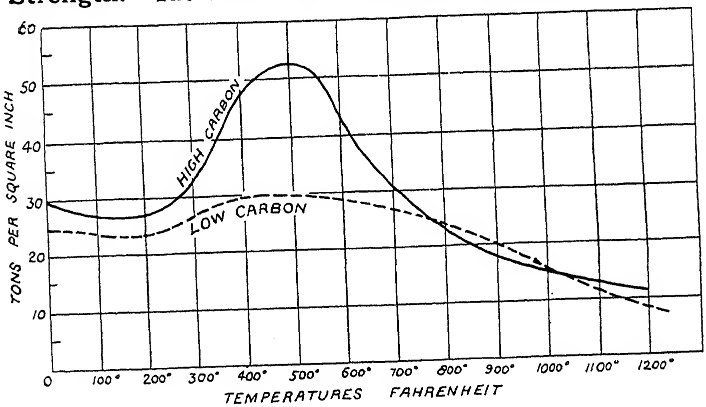


Fig. 954. Strength of Steel at High Temperatures.

is peculiar, and is indicated by the diagram in Fig. 954. At first the strength decreases slightly, but then steadily increases till it reaches a considerable maximum at about 500° F., then decreases

again till to a very low value at 1200°. This information is of the greatest possible value to the designer of boiler furnaces. Two examples have been given in the diagram, one shewing steel of high carbon composition, and the other of a milder quality, and the different results indicate clearly the effect of the presence of the carbon.

Pp. 393 and 840. **Chrome-Vanadium Steel.**—Under this title a new material has been introduced by Captain Sankey and Mr. Kent Smith. It consists of a carbon steel to which has been added small percentages of chromium and vanadium, and the result may be briefly stated as follows :—

Percentages added.	Yield Point tons sq. in.	Ultimate Stress tons sq. in.	Elongation on 2 ins. per cent.	Reduction of Area per cent.
Chromium 1% ... Vanadium ·15%	36·2	48·6	24	56·6
Chromium 1% ... Vanadium ·25%	49·4	60·4	18·5	46·3
Crucible Carbon Steel ...	16·0	27·0	35·0	60·0

The last line shews figures for the carbon steel without admixture, and the other lines indicate the effect of mixing percentages of chromium and vanadium to the steel. Higher strengths, even to 77 tons ultimate, were obtained, but the elongation then decreased to 13%.

P. 407. **Riveted Joints.**—A lap joint is essentially an unsatisfactory construction, and its real strength is a matter of some doubt. From an experiment made by Prof. Barr and mentioned in the Inst. of Mech. Engineers' *Proceedings*, Sept. 1901, Plate 164, it would appear that the actual strength of such a joint is only about 80% of the value as calculated on p. 407. For butt joints the usual method of calculation may be fully accepted.

Example 72.—Design a single-riveted lap joint for ½ in. plates, where the rivets and plate are of equally strong material, and the

three stresses f_t f_s and f_b have the relative values of 1, $\frac{3}{4}$, and 2. Bearing stress is to be taken into account.

The rivet shear, the plate tension, and the rivet bearing will have the respective strengths

$$\underset{(1)}{\frac{\pi d^2}{4} f_s} = \underset{(2)}{(p-d)\,t f_t} = \underset{(3)}{d t f_b}$$

Combining (1) and (3) we have

$$\frac{\pi d}{4} f_s = t f_b \qquad \therefore\ d = \frac{4t}{\pi} \times \frac{f_b}{f_s}$$

And by (2) and (3), substituting for d,

$$p\,t f_t - \frac{4t^2}{\pi}\frac{f_b}{f_s} f_t = \frac{4t^2}{\pi} \times \frac{f_b^2}{f_s}$$

$$\therefore\ p = \frac{4t}{\pi}\frac{f_b^2}{f_s f_t} + \frac{4t}{\pi} \times \frac{f_b}{f_s} = \frac{4t}{\pi}\frac{f_b}{f_s}\left(\frac{f_b}{f_t} + 1\right)$$

For $\frac{1}{2}''$ plates the values will be

$$d = \frac{4 \times 7}{2 \times 22} \times \frac{2 \times 4}{3} = \underline{1{\cdot}7}$$

$$p = \frac{4 \times 7}{2 \times 22} \times \frac{2 \times 4}{3} \times 3 = \underline{5{\cdot}1}$$

This problem is interesting from a theoretical standpoint, but the proportions could not be adopted, for practical reasons of manufacture, such as the difficulty of riveting and of procuring staunchness.

The following investigation is due to Prof. Pullen :

Let $d = 1{\cdot}3\sqrt{t}$ (as on p. 407), and let the ratio of f_b to $f_s = 2$. Comparing the strength of the rivet for shearing and bearing we have, in *Single Shear*,

$$\frac{\pi}{4} d^2 f_s = d t f_b \qquad \text{and } \frac{f_b}{f_s} = \frac{1{\cdot}02}{\sqrt{t}}$$

$$\therefore\ \text{When} \left\{ \begin{array}{lll} t = {\cdot}2 & \dfrac{f_b}{f_s} = 2{\cdot}28 \\ t = {\cdot}3 & \dfrac{f_b}{f} = 1{\cdot}86 \\ t = {\cdot}4 & \dfrac{f_b}{f_s} = 1{\cdot}62 \end{array} \right.$$

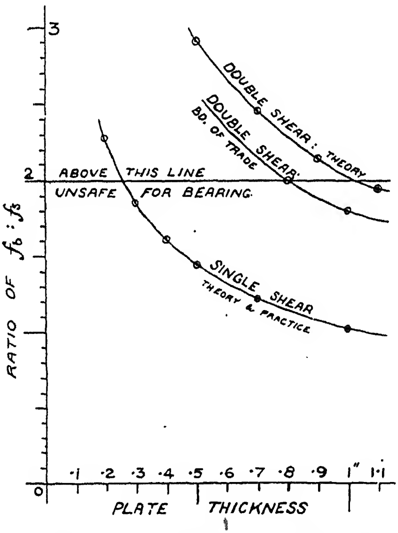
shewing that bearing surface is ample when t is $\frac{1}{4}$ in. and over.

In *Double Shear*

$$2\frac{\pi}{4}d^2 f_s = dtf_b \qquad \text{and} \quad \frac{f_b}{f_s} = \frac{2{\cdot}05}{\sqrt{t}}$$

$$\therefore \text{ When } \begin{cases} t = {\cdot}9 & \dfrac{f_b}{f_s} = 2{\cdot}16 \\ t = 1 & \dfrac{f_b}{f_s} = 2{\cdot}05 \\ t = 1{\cdot}1 & \dfrac{f_b}{f_s} = 1{\cdot}96 \end{cases}$$

So that below 1·05″ thickness the bearing stress is more than twice the shear. The full value of the shear is, however, rarely reckoned upon; indeed, the joint is usually treated as though it



Bearing Stress on Rivets.
Fig. 955.

were in single shear, though of a better construction than a lap joint. The Board of Trade adopt the number 1·75 as the ratio of $f_b = f_s$, but even then, as will be seen from the design in Fig. 955, all butt joints below ·8 in. plate must necessarily be treated as lap joints.

P. 428. Bending Theory.

Example 73. Make a diagram to shew how the moment of resistance to bending increases as the radius of curvature diminishes. Estimate the work done in bending a mild steel plate 1 in. thick, 5 ft. wide, and 20 ft. long, to a radius of 6 ft. (Machine Constr. Hons. Exam.)

$$\text{From p. 428:} \qquad B_m = fZ = \frac{EI}{\rho}$$

$$\therefore \text{ Moment of resistance, } \quad fZ \propto \frac{1}{\rho}$$

The diagram is given at G, Fig. 956. As the radius of curvature diminishes from a to a_1 the moment of resistance increases from b to b_1; and as $a \times b$ is constant, the curve is a rectangular hyperbola.

To estimate the work done, let the plate be bent from A B to

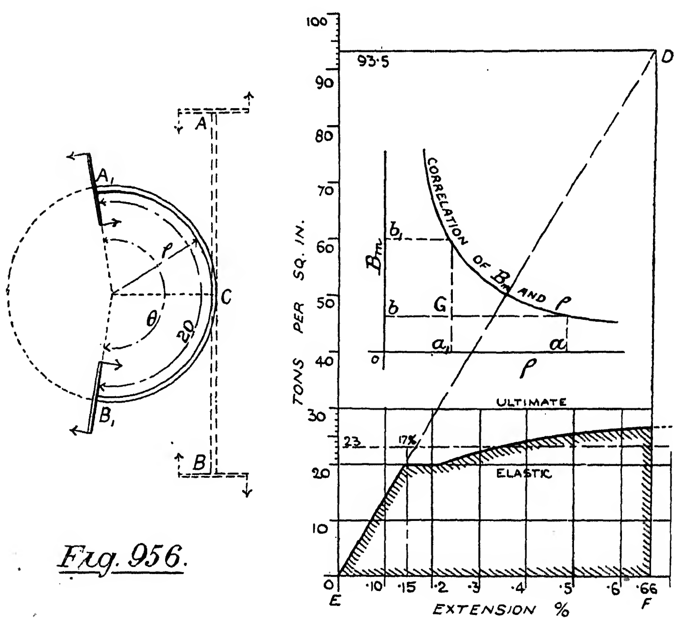


Fig. 956.

Work done in Bending.

A_1 B_1 by means of opposing couples that rotate relatively through the angle $\theta = \frac{20}{\rho}$. Each couple is—

$$B_m = \frac{EI}{\rho}, \qquad \text{or a force of } \frac{EI}{\rho^2} \text{ at an arm } \rho$$

This force is exerted, during rotation, through a space of 20 ft.

$$\text{Apparent work done} = \frac{EI}{\rho^2} \times 20 = \frac{Ebh^3}{\rho^2 12} \times 20$$

$$= \frac{30{,}000{,}000 \times 60 \times 1 \times 20}{72 \times 72 \times 12} = 558{,}000 \text{ foot pounds}$$

on the assumption that the stress and strain are within the elastic limit of the material. Examining further,

Total circumference at neutral axis $= 2 \times \pi \times 72 = 452{\cdot}55$

Total outside circumference $= 2 \times \pi \times 72\frac{1}{2} = 455{\cdot}55$

Extension on circumference $= 3$ ins.

Extension on 20 ft. length of plate $= \frac{240 \times 3}{452} = 1{\cdot}59$ ins.

Extension per cent. $= {\cdot}66$ (apparent).

Also stress in tons per sq. in.

$$= f = \frac{Ey}{\rho} = \frac{30{,}000{,}000 \times {\cdot}5}{6 \times 12 \times 2240} = 93{\cdot}5 \text{ (apparent).}$$

Taking co-ordinates, make EF $= {\cdot}66\,\%$ and DF $= 93{\cdot}5$ tons. Join E D, the elastic line. Refer now to diagram E, Fig. 345, p. 386, and plot it within the triangle DEF, Fig. 956, to form the shaded area. We now see that the apparent work estimate must be considerably modified, for the elastic tension cannot exceed ${\cdot}15\,\%$, and the real work will not be represented by the area DEF as was supposed, but by the shaded area. Now

$$\text{Area DEF} \propto \frac{93 \times 66}{2} \propto 3060$$

$$\text{Shaded area} \propto \frac{49 + 66}{2} \times 23 \propto 1320$$

$$\therefore \text{ Real work done} = \frac{1320 \times 588{,}000}{3060} = \underline{254{,}000 \text{ foot pounds.}}$$

P. 429. **Moment of Resistance by Calculation.** Referring to Fig. 957, let O_1 be the axis, at right angles to the paper, of the centroid of an area, and let a small element a of the area be placed at P. Let the 2nd moment round O be called I_1 and that round O_2 an axis parallel to that through O_1 be called

I_2, the value of which is required. The distance apart of these axes is x, so by Euclid II. 13

$$f^2 = b^2 + x^2 + 2xd,$$

$$\text{And } I_2 = \Sigma\, m \cdot f^2 = \Sigma\, m \cdot b^2 + \Sigma\, m \cdot x^2 + 2x\, \Sigma\, m \cdot d$$

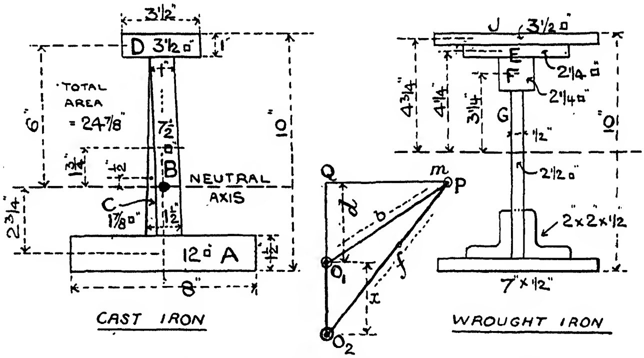
$$\text{But, having regard to sign, } \Sigma\, m \cdot d = \text{zero,}$$

$$\text{Also, } \Sigma\, m \cdot b^2 = I_1$$

$$\text{And, } \Sigma\, m \cdot x^2 = \text{whole area} \times x^2$$

$$\therefore I_2 = I_1 + (\text{area} \times x^2)$$

We may now find the 2nd moment of any beam section that



Moment of Resistance. Fig. 957.

can be cut up into a number of regular figures whose 2nd moments are obtainable by well-known formulæ, the total value being the addition of the moments of the parts round the neutral axis.

Two examples have been given in Fig. 957. Taking the **cast-iron beam**, the neutral axis is found by moments round H, as at top of p. 432.

$12 \times 0 = 0$		12
$7\frac{1}{2} \times 4\frac{1}{2} = 33{\cdot}5$		$7\frac{1}{2}$
$1\frac{7}{8} \times 3\frac{1}{4} = 4{\cdot}24$		$1\frac{7}{8}$
$3\frac{1}{2} \times 8\frac{3}{4} = 30{\cdot}6$		$3\frac{1}{2}$
Total moments $= 68{\cdot}34$	Total area $=$	$24\frac{7}{8}$

$$\text{Height of neutral axis from H} = \frac{68{\cdot}34}{24{\cdot}87} = 2{\cdot}75$$

Using the general formula $I_2 = \frac{bh^3}{12} + ax^2$

$$\text{2nd moment of D} = \frac{3{\cdot}5 \times 1}{12} + (3{\cdot}5 \times 36) = 126{\cdot}29$$

$$\text{,, ,, A} = \frac{8 \times 1{\cdot}5^3}{12} + (12 \times 2{\cdot}75) = 32{\cdot}44$$

$$\text{,, ,, B} = \frac{1 \times 7{\cdot}5^3}{12} + (7{\cdot}5 \times 1{\cdot}75^2) = 57{\cdot}8$$

$$\text{2nd moment of C} = \frac{bh^3}{36} + ax^2$$

$$= \frac{{\cdot}5 \times 7{\cdot}5^3}{36} + (1{\cdot}87 \times {\cdot}5^2) = 6{\cdot}26$$

$$\text{Total value of 2nd moment} = 222{\cdot}79$$

$$\therefore \text{Moment of resistance} = f_t \frac{I}{y_t}$$

$$= \frac{1{\cdot}25 \times 222{\cdot}8}{3{\cdot}5} = \underline{79 \text{ ton ins.}}$$

The **wrought-iron beam** has the neutral axis at its centre. We will therefore take the 2nd moment of J + E + F and double it, afterwards adding that of G.

$$\text{2nd moment of J} = \frac{7 \times {\cdot}5^3}{12} + (3{\cdot}5 \times 4{\cdot}75^2) = 79{\cdot}073$$

$$\text{,, ,, E} = \frac{4{\cdot}5 \times {\cdot}5^3}{12} + (2{\cdot}25 \times 4{\cdot}25^2) = 40{\cdot}547$$

$$\text{,, ,, F} = \frac{1{\cdot}5 \times 1{\cdot}5^3}{12} + (2{\cdot}25 \times 3{\cdot}25^2) = 24{\cdot}02$$

$$143{\cdot}640$$

$$\text{Doubled} = 287{\cdot}28$$

$$\text{2nd moment of G} = \frac{{\cdot}5 \times 5^3}{12} = 5{\cdot}2$$

$$\text{Total value of 2nd moment} = 292{\cdot}48$$

$$\therefore \text{Moment of resistance} = f_t \frac{I}{y}$$

$$= \frac{5 \times 292{\cdot}5}{5} = \underline{292 \text{ tons ins.}}$$

Pp. 431 and 848. **Measuring Areas.**—The Amsler Planimeter is now well known as a most useful instrument for the mechanical integration of areas, and is shewn in Fig. 958. A

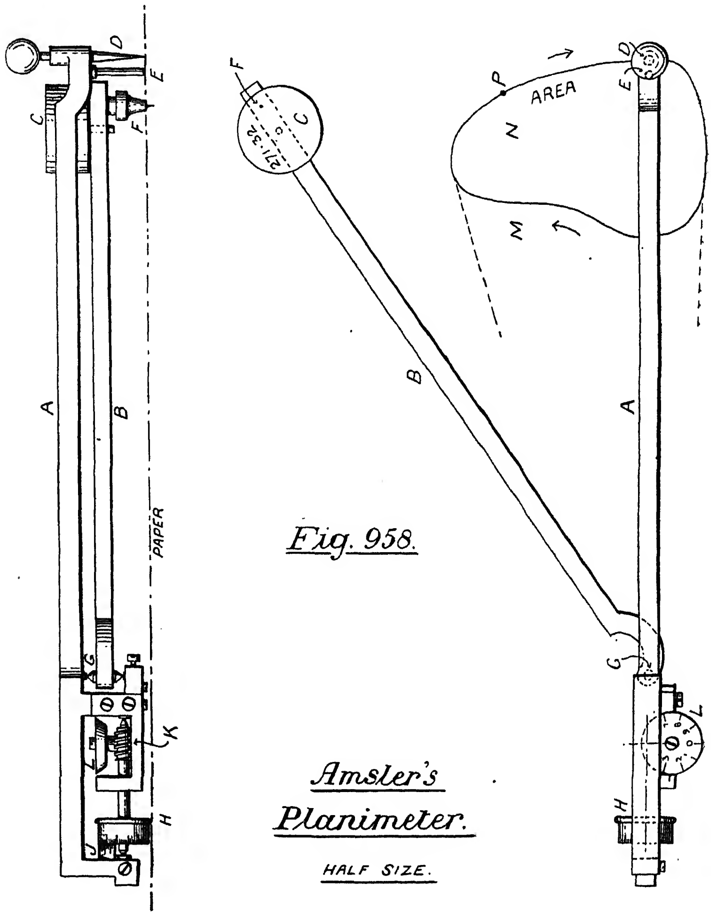


Fig. 958.

Amsler's Planimeter.

HALF SIZE.

bar B is pivoted at one end on a stud F that is held to the paper by a pin, and a weight C laid upon the bar prevents lifting. A second bar A is hinged to B at G, and is supported by the wheel H

and stud E. The latter is just so long as to keep the pointer D touching but not indenting the paper. If an area be traced out by the pointer D, the wheel H will roll on the paper, sometimes forward and sometimes backward, but the net result will be forward and will indicate on the vernier J, say in square inches, the exact value of the area. If the net revolutions of H are more than one, the observer must look at the dial L, where the turns of H are registered by the worm wheel K, without the tedium of personal counting. The theory of the instrument is very simple, the wheel merely registering the difference of two total radial areas N and M; but this knowledge is unimportant.

To use the instrument, let the tracer D be set to some point P on the boundary of the area; and note the readings on the wheels L and H, and vernier J, as follow:

Reading on wheel L	=	2 +	=	20·00
" " H	=	4·7 +	=	4·70
Coincidence on vernier J	=	2	=	2
Correct reading			=	24·72

Now trace the boundary by means of pointer D, in the direction of the arrow till point P is again reached, and take the reading of the wheels again as, say, 29·65. The actual area will be 29·65 − 24·72 = 4·93 sq. ins.

A special case occurs when the area is so large that it can only be circumscribed by placing the pin point F *within* the diagram. The procedure is not changed in the reading, tracing, and re-reading; but the result must be increased by a certain constant, which varies with different instruments and is marked upon the weight C. For example, let

Reading after tracing	=	45·37	
" before "	=	34·38	
Difference	=	10·99 ±	
Add figure on weight	=	271·32	
Area	=	282·31 or 260·33 sq. ins.	

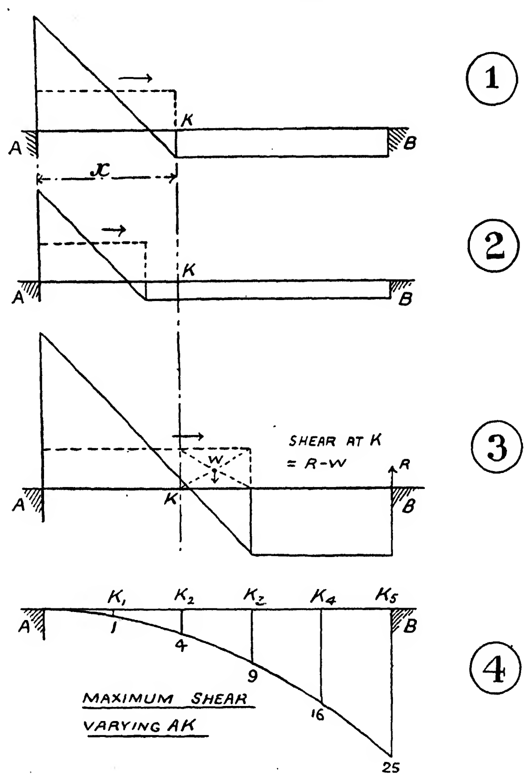
N.B.—'Difference' is positive or negative, as found by rough trial of the area.

The Planimeter is immensely useful for a variety of area

measurements, such as indicator diagrams, the stress diagrams of beam sections, &c. &c.: *see* pp. 431–436, 623, 679, 846.

P. 444. Rolling Load.

Example 74. Prove an algebraic formula to shew that with a continuous load of uniform intensity passing over a beam A B, such as when a long train passes over a bridge A to B, the maximum shearing stress to any point K of the beam occurs when the part A K is fully loaded while the part K B is entirely unloaded; and that the magnitude of the stress is proportional to the square of the distance of K from the point A. (Hons. Applied Mechanics Exam.)



Rolling Load Problem. Fig. 959.

Referring to Fig. 959. Let a load of w lbs. per foot run be advanced from the left up to any point on the beam. It is required to find the shearing force at the section K.

Diagram (1). The load is advanced up to K. Reaction at A has the general form $wx\left(l - \frac{x}{2}\right) \div l$ for every case. Reaction at B, which is shear at K is $wx \cdot \frac{x}{2} \div l$.

Diagram (2). The load is short of K. The shear at K is less than before, because both reactions are less.

Diagram (3). When loaded beyond K. Both reactions are increased ; but the shear at K is actually less than in (1) for *all* the load beyond K has to be deducted in arriving at the shearing force, whereas only *part* of it adds to the reaction.

Therefore the greatest shear at K occurs when the load travels exactly up to that point.

Next take various positions for K between A and B. The ordinates of the curve of maximum shear for these points will be, from diagram (2) $= wx \times \frac{x}{2} \div l = \frac{wx^2}{2l}$

But x is the abscissa from A

$\therefore$ Ordinates $\propto x^2$

and the curve is drawn at (2).

Pp. 446, 464, 470, and 885. **Link Polygons.**—It is required to add, that is find, the resultant of a number of forces inclined at various angles, but whose directions are co-planar, that is, lie within the same plane. The forces may meet in one point, and the method on p. 464 be found convenient; but if the directions are varied in any possible co-planar manner, the construction known as the link or funicular polygon must be adopted.

Let the lines A B C D E, Fig. 960, represent forces in direction, sense, and magnitude whose resultant is required. Draw the unclosed polygon 0 1 2 3 4 5 by lines parallel to the forces, whose lengths are also proportional to their magnitudes. The closing line R will naturally shew the magnitude and direction of the resultant, and the sense will be reversed, as shewn by the arrow. We must next shew on the space diagram the exact position of R. Choose any pole O in the vector figure, either in or out of the

polygon, and join lines to corners. Commencing at any point on A draw the polygonal lines shewn in space diagram parallel to the radials in vector figure and in order, and letter as shewn. These lines are known as the links, and are lettered according to Bow's method. O o_1 and O 5_1 are the first and last links respectively, and at their intersection x is placed the resultant force line R, parallel to o 5. The proof is very simple. Imagine the space polygon to be a framed structure or endless string held taut by

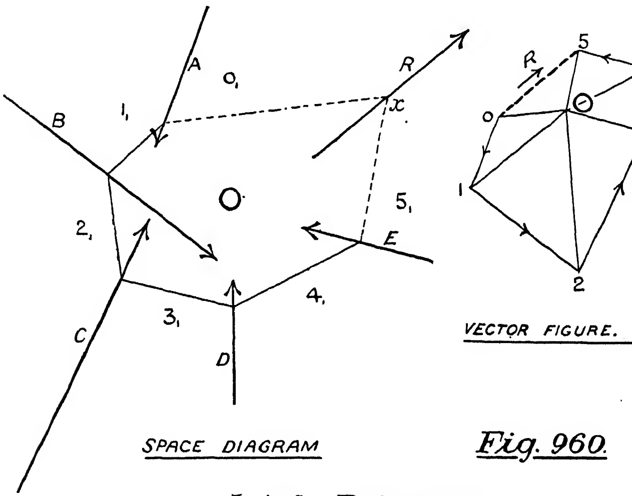


Fig. 960.

Link Polygons.

the original forces. Every corner of this string has three forces, as for example $O_1 1$ $1_1 O$ $O o_1$: and these have the force diagram O o 1_1. It follows that the whole of the vector figure is the force diagram for the framed structure, and if R's direction be reversed all the forces are in balance.

Two particular cases occur: (1) When the vector figure *only* is closed, o falling on 5, the resultant is a couple whose forces are each equal to O o or O 5, and whose area will be found from the space diagram; (2) when both polygons close, the resultant is zero, the forces being already balanced.

Change of Pole. In Fig. 961 are shewn six forces exactly balancing, as proven from the vector figure. Let the link polygon be drawn in the first place for a pole O, and let it be desired to re-draw it for a new pole O_1. This has been done point by point in the usual manner, arriving at the dotted polygon, but could have been effected more simply by the use of a simple property connected with two poles of the same vector figure. Join O O_1 producing both ways. Next take any line of one link polygon and the corresponding line on the other link polygon; produce these till they meet, obtaining, say, the point e_3. Doing this for all the corresponding sets of lines of the polygons, we have the additional points e_1 e_0 e_4. It will now be found that these points are in one straight line, that is itself parallel to the lines O O_1. For proof:

$$o1 = oO + O1 \qquad \text{also, } o1 = oO_1 + O_11$$

But because o1 balances 1o,

$$oO + O1 \text{ balances } O_1o + 1O_1$$

$$\text{and } oO + O1 + O_1o + 1O_1 = \text{zero}$$

$$\text{But } O_1o + oO = O\,O^1$$

$$\text{and } O1 + 1O_1 = O_1O$$

The force O O_1 is therefore the resultant of the corresponding link pairs in either link polygon. Its reciprocal e_1 e_0 must join the meets of the link pairs, and must be parallel to O O_1. A similar treatment of other pairs will prove the line e_3 to e_4 also parallel to O O_1.

Use of Link Polygon.—As an example take a semicircular roof, of which half is shewn at A B C, Fig. 962. Let the wind blow from right to left, and we require to know the direction and magnitude of the resultant of all the pressure that comes upon C B, in order to carry out the construction given at p. 470, and thus arrive at the stresses on the members. Divide the arc B C into any convenient number of equal parts at D, E, F, G, H, J, K, L, in this case nine. Assume the pressure intensity uniform over each division, though varying from one division to another, and let each piece of area acted upon be $h' \times b'$, where h' is the height and b' the breadth between the bays, both in feet. If the wind intensity be w, the normal intensity n will vary from w at B to zero at C, and its value at any point can be found from the

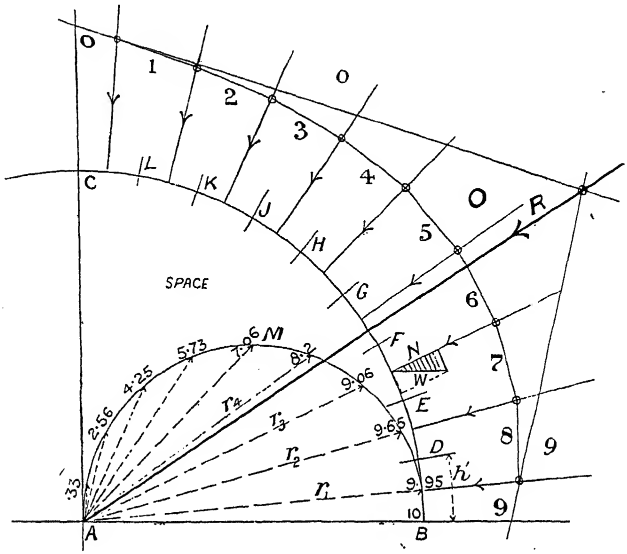
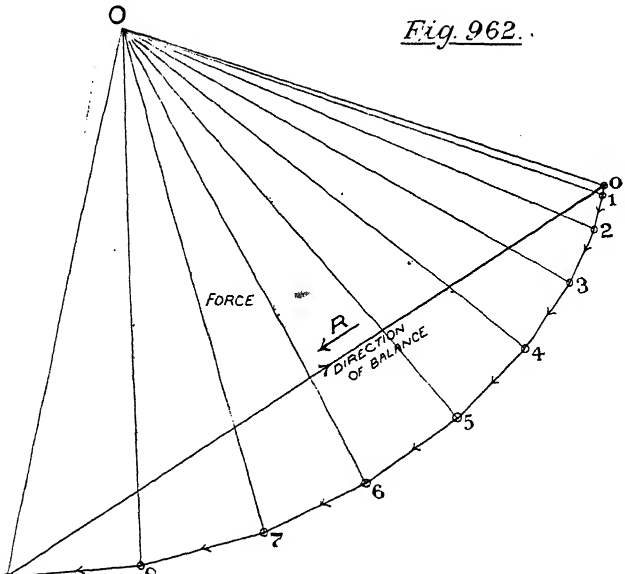


Fig. 962.



Wind Pressure on Roof.

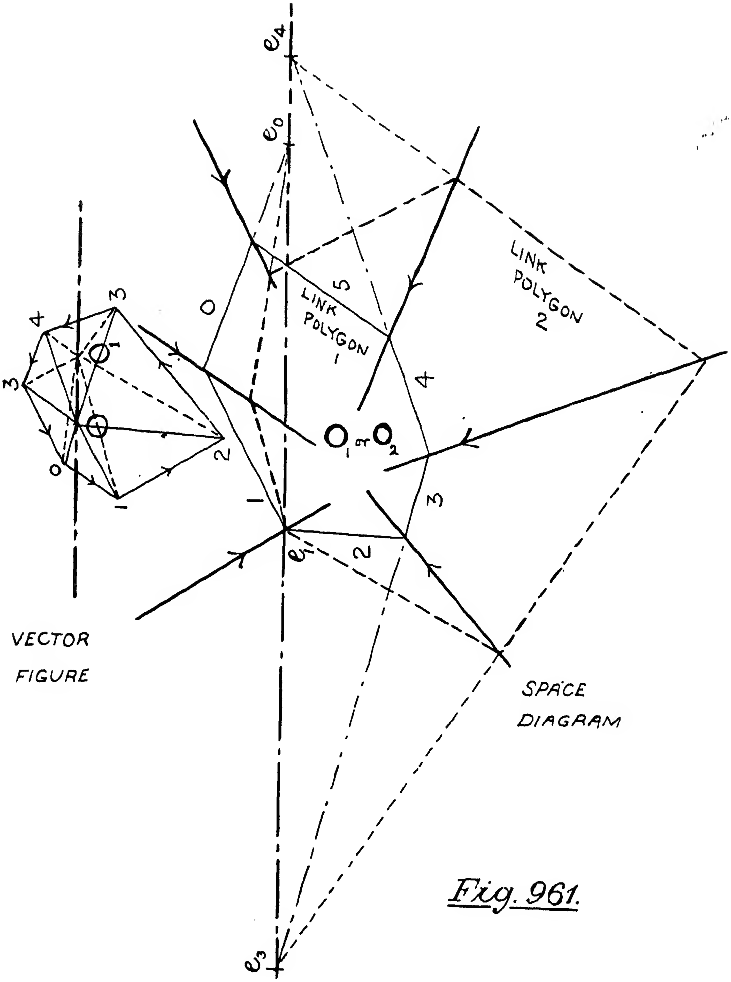


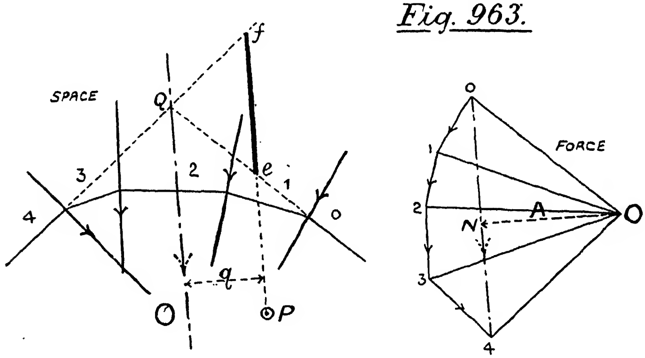
Fig. 961.

Rotation in Link Polygons.

diagram A M B. The line A B is w say, or 56 lbs. per sq. ft.; then the dotted radii, r_1 r_2 r_3 &c., will shew the value of n for the respective divisions. The total force on each area being represented by an arrow, its magnitude will be $n\,h'\,b'$, or $56 \times \frac{r}{10} \times h'\,b'$. These values are next set out in the vector figure below on lines parallel to the forces taken in order, as 0 1, 1 2, 2 3, 3 4, &c., and 0 9 will be the magnitude of the resultant which passes through the intersection of the first and last link in the link polygon, and also through the centre A of the roof.

Moments in Link Polygons.—Four forces in Fig. 963 are treated in the usual manner, and a vector figure obtained where the pole O has been placed at a known perpendicular distance A

Fig. 963.



Moments in Link Polygons.

rom the resultant 0 4. Nothing but distances should be measured on the space diagram, and forces only on the vector figure. In the former the resultant is placed at Q, where the first and last links intersect. Let it now be required to find the moment of the resultant round the point P, and draw line P $e\,f$ parallel to the direction of the resultant. Naturally the moment will be $0\,4 \times q$. But it is also O N $\times\, e\,f$ where $e\,f$ is the intercept between the first and last links. For proof, 0 O 4 and Q $f\,e$ are similar triangles: hence—

$$\frac{ef}{0\,4} = \frac{q}{\mathrm{O\,N}} \quad \text{and} \quad 0\,4 \times q = \mathrm{O\,N} \times ef.$$

The product of polar distance by intercept is an easier method of finding a moment, because the former can be fixed artificially as 10, or some simple multiple thereof.

This proof will also serve for the Culmann bending moment diagram, pp. 446 and 885.

Problem. To draw a link polygon which shall pass through three given points in the space diagram.

As the practical application is to cases of vertical and parallel forces, such case only will be here considered. Re-

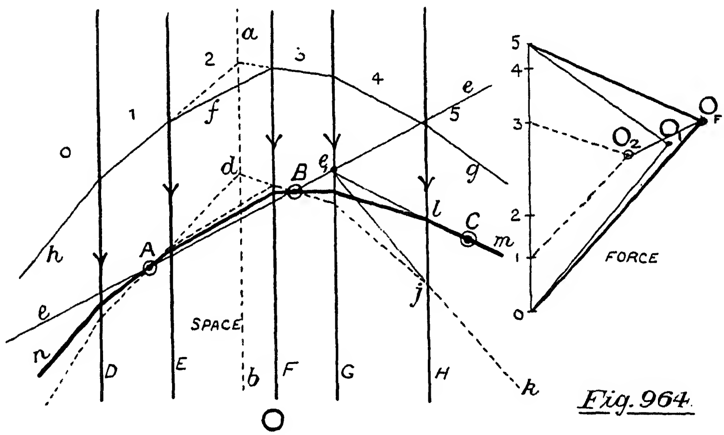


Fig. 964.

Link Polygon through 3 given points.

ferring to Fig. 964: let A B C be the given points, and D, E, F, G, and H the given forces. The latter are set down in magnitude and direction as 0 1, 1 2, 2 3, 3 4, and 4 5 on the vector figure.

1. Firstly, make no attempt to pass through the points, but choose any pole O_1 and draw the polygon *h f g*.

2. Take the two links O_1 and O_3 immediately over A and B and produce them to find the resultant *a b*. The same pair of links in any other polygon would have the same resultant, for the pole can be chosen anywhere.

3. Take any point *d*, join to A and B and produce, forming two such links in a new polygon.

4. Draw parallel lines in the vector figure, shewn dotted, and obtain the pole O_2 for a polygon passing through A and B only.

5. To pass through A, B and C, adopt the rotation method, where A and B will be turning points, and the line *e e* a line of turning points.

6. Draw $O_2 O_F \parallel$ A B: also produce *j k* to e_1 and join e_1 to C; then draw $5\ O_F \parallel l\,m$, and the final pole O_F is found.

7. The link polygon through A B C may now be completed by rotation, or by reference to the vector figure $5 O_F o$.

Applications of the foregoing problem. Let us first consider the conditions as to the distribution of pressure between the voussoirs or blocks of a stone arch. In Fig. 965, V V are a pair of voussoirs, held together by mortar that may be sufficient to

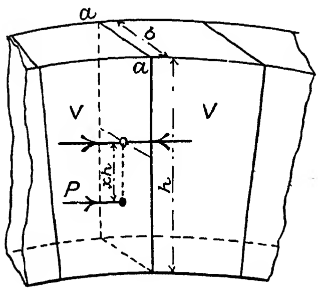


Fig. 965.

Pressures on a Stone Arch.

resist sliding, but which is not to be subjected to any tension whatever. So long as the direction of pressure is in the centre of the voussoir we feel sure that the conditions are safe; but we now require to know how far the line of pressure may deviate above or below the centre line without causing the joint to open. Let the pressure be exerted by the force P, whose distance downward from the centre is *xh*. We may place two balanced forces at the centre line, each equal to P, without altering the conditions, and we now have two actions to consider: (1) a moment P × *xh* tending to open the joint at *a a*, and (2) a pressure P at the centre line, tending to close the joint. If these two actions balance, the joint will have neither pressure or tension, but will remain closed.

Then, to open:

$$P \times xh = f\frac{bh^2}{6} \qquad \text{and the opening stress } f = \frac{6\,Px}{bh}$$

And to close:

$$P = fbh \qquad \text{closing stress being } f = \frac{P}{bh}$$

But these must balance:

$$\therefore \frac{6\,Px}{bh} = \frac{P}{bh} \quad \text{and } x = \frac{1}{6}$$

so P cannot be placed further from the centre than $\frac{1}{6}h$, or the line of pressure must be kept within the *middle third* of the voussoir depth *h*.

The **Masonry Arch**, Fig. 966, is drawn in half only, the loads being symmetrically disposed; and there are five voussoirs, *v v* &c., to each half. The loads *w w* &c., are found by erecting perpendiculars *a b c d e f*, each load being the resultant of the weight of wall between a pair of verticals combined with the weight of the voussoir, the latter being, of course, proportionately small. Set these loads out in the force diagram as 0 1, 1 2, 2 3, &c. By symmetry the force O o must be horizontal. Take any pole O_1 and draw the link polygon *g h*, then find *r s* the resultant of Oo and O4 the pressures at skewback and key respectively. Take points *l* and *k* and find a new polygon passing through them, such that the link O o still remains horizontal and meeting *r s* in *t*. Join *t k* and draw the line 4 $O_2 \parallel t\,k$, thus giving the new pole O_2. Complete the polygon *k l*, which will indicate the line of pressure or resistance between the voussoirs, the magnitudes of the forces being found from the force diagram. The line *k l* must lie within the middle third of the joint, as shewn dotted, and if this does not occur everywhere a readjustment must be made. The extreme possible conditions are shewn at E and F, where the line *m n* is better than *p q*, because it causes a lower value for the pressures. The greatest pressure occurs at the skewback or support, and the intensity must never be more than the material will allow. If the line of resistance passes through the centre the intensity is considered uniform, but if the line touches the dotted arc the maximum intensity will be twice that of the mean.

A Three-pin Bridge, Fig. 967, consists of two cantilevers *l* C and A *f*, springing from abutments *l* and *f*, and the middle portion is completed by the spans C B and B A resting on pins at C, B, and A. Sometimes the centre spans consist of drawbridges or bascules, and in other cases are unlifted; but the construction

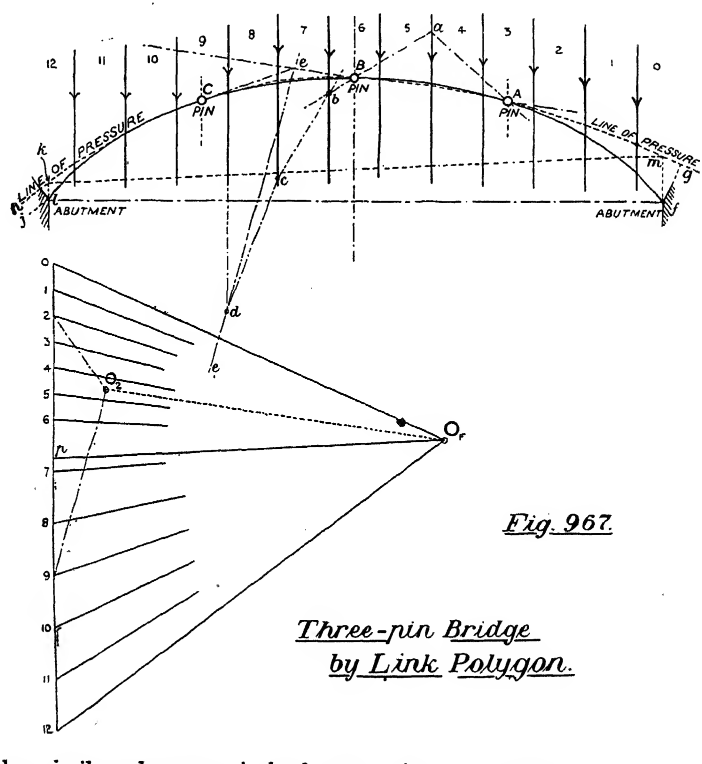
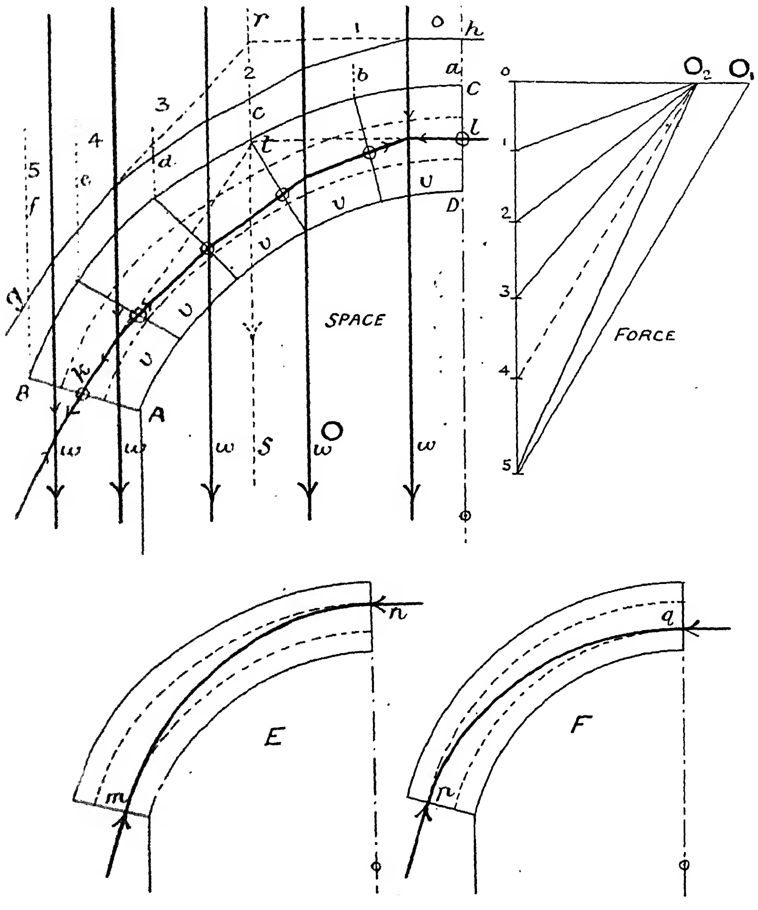


Fig. 967.

Three-pin Bridge by Link Polygon.

has similar advantages in both cases, viz., the bending moment is almost eliminated, and certainly has a zero value at each pin. The loads being shewn by vertical lines on the space diagram and by their values in the force diagram, it is required to draw a link polygon through the three pins, and the deviation of the links from the curved line of the bridge will indicate the bending moment. The loads have been imagined smaller on the left than



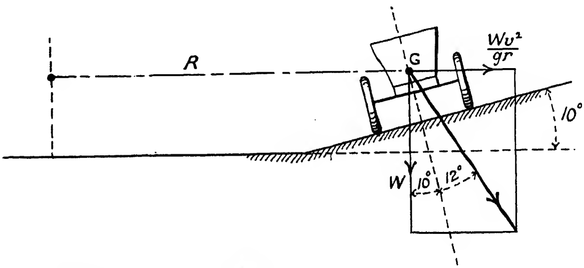
Masonry Arches *Fig. 966.*

on the right side, but all equal in value on one side. The preliminary polygon for O_1 pole is not required, for the line of resultant for the links O 3 and O 6 is evidently passing through point a. Join this point to A and B and find the pole O_2 for a polygon passing through A and B. Next join A B and produce leftward. This is the line of rotation. Produce line $d\,e$, the link O 9 till it meets A B produced in e_1; and join C e_1 giving the link O_9 in the final polygon. Draw a line in force diagram || C e_1 from point 9, and another line $O_2\,O_F$ || A B. The intersection of these lines is O_F the pole of the final polygon, which may now be completed in the space diagram, and which will, of course, pass through the three points A, B, and C. This is the line of pressures, and coincides very nearly with the arc of the bridge from C to A, It deviates appreciably at the abutments, and causes a bending mount on the left of O 12 × $h\,j$, but a larger moment of O 0 × $f\,g$ on the right. The thrust between the abutments is nearly horizontal and is found by joining $k\,m$ in the space diagram. The parallel p O_F in the force diagram gives the value.

CHAPTER IX.

P. 474. **Velocities:**

Example 75. A motor-car moves in a horizontal circle of 300 ft. radius. The track slopes sideways at an angle of 10° with the horizontal. A plumb-line on the car makes an angle of 12° with a



Velocity of Motor-Car. *Fig. 968.*

perpendicular to the track. Find (1) the speed of the car, (2) the coefficient of friction if the car is just on the point of side-slipping. (Board of Education Exam. Stage 3, 1904.)

Referring to Fig. 968. There are two forces, the weight of the car and the centrifugal force, and the resultant of these takes the same direction as the plumb-bob string, for their ratio is independent of weight, viz. :—

$$\frac{Wv^2}{gR} \div W = \frac{v^2}{gR} = \tan 22^\circ$$

$$\therefore v^2 = \tan 22^\circ \times gR$$

$$\text{And } v = \sqrt{.404 \times 32 \times 300} = 62{\cdot}2 \text{ ft. per sec.}$$

$$\text{or } \underline{42{\cdot}4 \text{ miles per hr.}}$$

Also the angle of resultant with the track = 12°

$$\therefore \mu = \tan 12^\circ = \underline{{\cdot}208.}$$

Example 76. A Tramcar weighs 50 tons, and current is cut off when the speed is 16 miles per hr. Reckoning time from that instant, the following velocities and times were noted:—

V (miles per hr.)	16	14	12	10
t (time in secs.)	0	9·3	21	35

Calculate the average retarding force, and find the average velocity from $t = 0$ to $t = 35$. Find distance travelled between these times.

If the law of resistance be F lbs. $= a + bV + cV^2$, find the values of a, b, and c, from the above observations. (Hons. Applied Mech. Exam.)

Set out the velocities on a time base, Fig. 968*a*.

$$\text{Average velocity} = \frac{\text{area under V curve}}{\text{total time}} = \frac{129{\cdot}5 + 152{\cdot}1 + 154}{35}$$

$$= \underline{12{\cdot}44} \text{ miles per hr.}$$

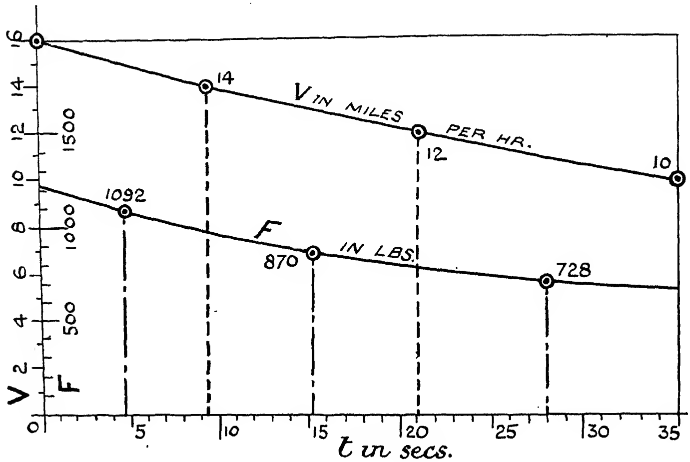
$$w = 50 \times 2240 \qquad f = \frac{v}{t} = \frac{V \times 5280}{t\ 3600} = 1{\cdot}46\frac{V}{t}$$

$$\text{F lbs.} = \frac{w}{g}f = \frac{50 \times 2240 \times 1.46}{32} \cdot \frac{V}{t} = 5100\frac{V}{t}$$

where V is the decrease of velocity in miles per hr.

$$\text{Hence average force} = \frac{5100 \times 6}{35} = \underline{874 \text{ lbs.}}$$

During time $9{\cdot}3 - 0 = 9{\cdot}3$	$F_1 = \dfrac{5100 \times 2}{9{\cdot}3}$	$= 1092$
" " $21 - 9{\cdot}3 = 11{\cdot}7$	$F_2 = \dfrac{5100 \times 2}{11{\cdot}7}$	$= 870$
" " $35 - 21 = 14$	$F_3 = \dfrac{5100 \times 2}{14}$	$= {\cdot}728$



Retardation Problem. Fig. 968a.

Set these up in the centre of the spaces to form a force curve.

To find distance, $s = tv$ for thin rectangles under the velocity curve, and the summation of v regarding t will be the distance required.

Distance travelled $= 35 \times 12{\cdot}44 \times 1{\cdot}46 =$ 636 feet.

Again,

$$F = a + Vb + V^2c$$

$$1092 = a + 15b + 225c \quad \text{.........} \quad (1)$$

$$870 = a + 13b + 169c \quad \text{.........} \quad (2)$$

$$728 = a + 11b + 121c \quad \text{.........} \quad (3)$$

Subtracting (2) from (1), and (3) from (2),

$$222 = 2b + 56c \quad \text{.........} \quad (4)$$

$$142 = 2b + 48c$$

$$\text{Subtract} \quad 80 = 8c$$

And $c = 10$

From (4) .... $b = -169$

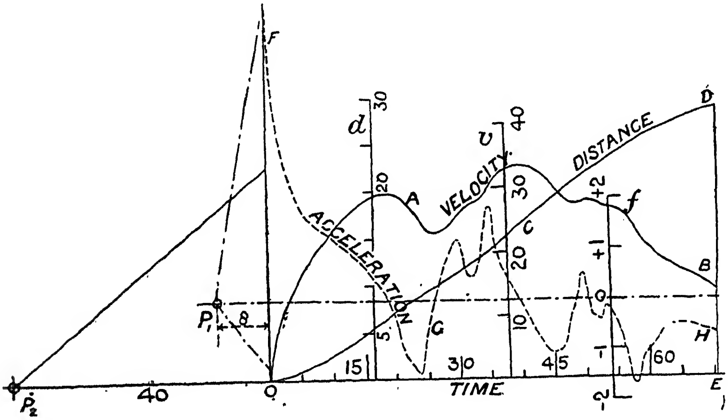
Lastly by substituting in (1) (2) or (3),

$a = 1377$

$\therefore$ F lbs. $= \underline{1377 - 169V + 10V^2}$

which is the equation to the force curve.

P. 474. **Velocity Curves on Time Base.**—The curve OAB, Fig. 969, represents by its ordinates the actual velocities of a locomotive, from the moment O of leaving the station till the



Velocity Curve on Time Base. *Fig. 969.*

expiration of a period of time OE. The scale v shews miles per hour, and the base scale the minutes from O. A polar distance OP_2 of 40 time units is taken, and the sum curve OCD is drawn by the method at p. 851. The ordinates of this curve, measured by the scale d, will shew the distances travelled from starting, for $d = tv$ is true when very small time values are taken at each step. Also $v = ft$, and f is the differential of v regarding t. Therefore the acceleration curve FGH is obtained as the slope of the velocity curve, using the polar distance at P_1 of 8 time units,

and adopting the method of p. 852. The scale f measures the accelerative value, and the correlation of all the scales will be understood from p. 853.

Pp. 478 and 680. **Energy of Rotation.**—At the former of these pages the energy of a rotating body was shewn to be

$$\cdot 0001714\ wR^2N^2 \text{ foot pounds} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where R = average radius of rotation *in feet*, or radius of gyration. Now 'revolutions per minute' is only a practical way of stating angular velocity:

$$\therefore \text{ Energy of rotation } = \frac{wv^2}{2g} = \frac{w\omega^2R^2}{2g} \text{ foot pounds.}$$

Taking I as moment of inertia of the body in inch lb. units, and m a small element of mass acting at a radius r_1 inches, we have

$$I = \Sigma\, mr_1^2 = \frac{w}{g} R^2 \cdot 12^2 \text{ in gravity units}$$

$$\text{or} = w\, R^2 \cdot 12^2 \text{ in absolute units.}$$

The second is the usual method of expressing I, and will be here adopted. Then—

$$\text{Energy of rotation} = \frac{w\, R^2\omega^2}{2g} = \frac{I\, \omega^2}{12^2 \cdot 2g} \text{ foot pounds} \ldots \quad (2)$$

$$\text{And from (1) this also} = \cdot 0001714 \frac{I}{12^2} N^2 \text{ foot pounds} \quad (3)$$

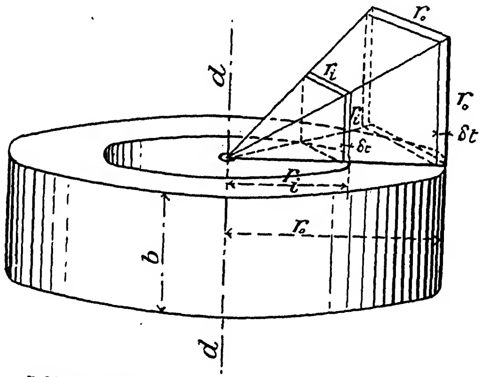
Note carefully that while I is generally measured in inches and lbs. it must be changed to feet and lbs. when inserting in the energy formula, or $w\, r^2$ becomes $w\, R^2 \times 12^2$, and $w\, R^2 = I \div 12^2$. We see then there are two methods of expressing the energy, one in N and one in ω.

Graphically.—The value of r^2 (radius of gyration squared) being known, that of I can be obtained. The graphic solution for r^2 for *any* solid of revolution is explained at pp. 681 and 845; but for certain regular solids, such as cylinders and rings, we may proceed by a method similar to that on p. 419. In Fig. 970 is shewn a **cylinder** of radius r_0 whose I is required round the

axis *d d*. Dividing the whole solid into thin rings, w_1 being the weight of a cubic inch in lbs. :

$$\begin{aligned} I &= \Sigma\,(\text{ring wt.} \times \text{ring radius}^2) \\ &= \Sigma\,(w_1 \times \text{vol.} \times r_1^2) \\ &= \Sigma\,(2\pi\, r\, b \cdot \delta t \cdot w_1\, r_1^2) \\ &= 2\pi\, b\, w_1\, \Sigma\,(\delta t \cdot r_1^3) \end{aligned}$$

But $\delta t \cdot r_1^3$ is the contents of any square lamina of the pyramid shewn, multiplied by its respective radius. Therefore $\Sigma\,(\delta t \cdot r_1^3)$



Moment of Inertia. Fig. 970.

will be the volume of the pyramid multiplied by the arm from centre *d* to the centre of gravity of pyramid.

$$\begin{aligned} \text{And } I &= 2\pi\, b\, w_1\, (\text{vol. of pyramid})\, \tfrac{3}{4}\, r_o \\ &= 2\pi\, b\, w_1\, (r_o^2 \times \tfrac{1}{3}\, r_o)\, \tfrac{3}{4}\, r_o \\ &= w_1\, b\, \frac{\pi}{2}\, r_o^4 \end{aligned}$$

The total weight of the cylinder being $w_1 \pi\, r^2\, b$, we have

$$r^2 = I \div \text{weight} = \frac{r_o^2}{2} \text{ inches.}$$

Referring again to Fig. 970, let the radii r_o and r_i represent

the outer and inner diameters respectively of a **ring** of width b. The Moment of Inertia of the ring will be found by deducting the I of the cylinder r_i from that of the cylinder r_o: so from previous reasoning,

$$\text{I of ring} = w_1 b \frac{\pi}{2} (r_o^4 - r_i^4)$$

and, dividing by the weight $w_1 \pi b (r_o^2 - r_i^2)$

$$r^2 = \frac{r_o^2 + r_i^2}{2}$$

MOMENTS OF INERTIA OF SOLIDS OF REVOLUTION.

Solid.	w $= w_1 \times$ vol.	r^2	r	I inch lb. units.
Cylinder of radius r_o	$w_1 \pi r_o^2 b$	$\frac{r_o^2}{2}$	$\frac{r_o}{\sqrt{2}}$	$\frac{w_1 \pi b r_o^4}{2}$
Ring of radii — r_o outside r_i inside	$w_1 \pi b (r_o^2 - r_i^2)$	$\frac{r_o^2 + r_i^2}{2}$	$\frac{\sqrt{r_o^2 + r_i^2}}{\sqrt{2}}$	$\frac{w_1 \pi b (r_o^4 - r_i^4)}{2}$

The moment of inertia of any fly-wheel can now be calculated by adding the moments of the separate parts; and the energy of rotation is deduced by reference to equations (2) or (3).

Experimentally.—The Energy of Rotation may also be found by direct experiment on the rotating solid. A fly-wheel A, Fig. 971, is fastened by a plate B to the ball-bearing hub C of a bicycle, and is supported by the bracket D, held to the wall by bolts. A small weight E is attached to a string wound round the hub C, and serves by its fall to cause rotation of the fly-wheel. Let the wheel be accurately balanced, so that it will rest equally well in any position, and let the frictional resistance of the ball-bearing be quite inappreciable.

The weight being w, the energy of the fall is wH foot pounds,

where H is the height of c from the ground. If the fall were unresisted, all the weight energy would be lost in impact, but as it is, some of the energy is given to the wheel, while the portion lost by impact will be calculated from the weight's velocity. Consider the velocity of weight and the angular velocity of wheel

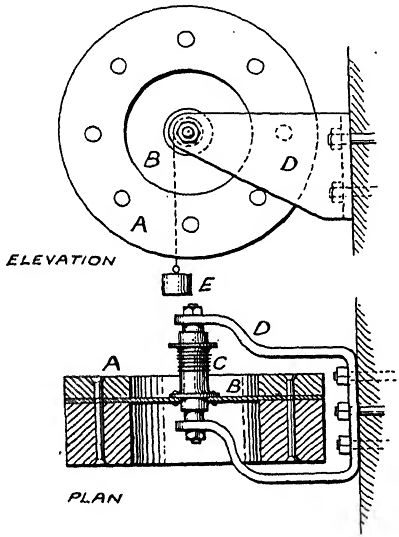


Fig. 971.

Energy of Rotation.

to be uniformly accelerated. Let t = time of fall in seconds, and n_1 the number of revolutions during fall. Then

$$\text{Average } n = \frac{n_1}{t} \text{ revs. per sec.}$$

$$\text{and Highest } \underline{n} = \frac{2n_1}{t} \text{ revs. per sec.}$$

$$\text{Max. weight velocity } v = 2\pi R\underline{n} = \frac{2\pi R\, 2n_1}{t} = \frac{4\pi}{12} \cdot \frac{rn_1}{t}$$

where r = hub radius in ins.

The kinetic energy in w when reaching ground $= \dfrac{wv^2}{2g}$: and the actual energy accumulated in fly-wheel at its highest speed will be wH less this kinetic energy, or

$$\text{Energy of rotation (by experiment)} = w\left(H - \frac{v^2}{2g}\right)$$

$$= w\left\{H - \left(\frac{4\pi}{12} \cdot \frac{rn_1}{t}\right)^2 \div 2g\right\} = w\left(H - \cdot 017 \frac{r^2 n_1^2}{t^2}\right) .$$

Next take careful measurements of fly-wheel section to find I in ins. and lbs., deducing

$$\text{Energy of rotation (by calculation)} = \cdot 0001714 \frac{I}{12^2} \cdot \frac{n_1^2}{60^2} \text{ foot pounds,}$$

and the two results ought to agree very closely.

M *of Fly Wheel.*—Professor Perry has given the name '**M** of a fly-wheel' to a constant belonging to each particular fly-wheel, such that MN^2 = energy in foot pounds. Hence, if the energy be divided by N^2, the value of **M** is found. The method is very convenient, for conversely the energy may be at any time calculated if the speed of revolution be known. Also

$$\cdot 0001714 \frac{I}{12^2} \cdot N^2 = MN^2$$

$$\therefore M = \cdot 000{,}001{,}19\, I = 119\, I \times 10^{-8}$$

$$\text{For a cylinder, } M = \frac{w_1 b r_o^4}{534{,}800}$$

$$\text{and for a ring, } M = \frac{w_1 b}{534{,}800}(r_o^4 - r_i^4)$$

$$\left.\begin{array}{r}\text{while the fly-wheel}\\ \text{experiment gives}\end{array}\right\} \quad M = w\left(H - \cdot 017 \frac{r^2 n_1^2}{t^2}\right) \div \frac{n_1^2}{60^2}$$

Example 77.—A Fly-wheel is required to store 12,000 foot pounds of energy as its speed increases from 98 to 102 revs. per m. What is its I?

The wheel being a solid disc of cast iron whose thickness is $\frac{1}{10}$ of its diameter, find D. (Hons. Applied Mechs. Exam.)

$$\text{Kinetic energy} = \cdot 0001714\, I\,(N_1^2 - N_2^2)$$

$$= \cdot 0001714\,(102^2 - 98^2)\, I = 12{,}000$$

$$I = wR^2 = \frac{12{,}000}{800 \times \cdot 0001714} = \underline{87{,}514} \text{ in absolute units (feet and lbs.)}$$

Further, cast iron weighing 450 lbs. per cub. ft.,

$$\text{wt. of fly-wheel} \ldots w = \frac{\pi}{4}D^2 \times \frac{D}{10} \times 450$$

$$= \frac{22 \times 450}{7 \times 4 \times 10} \times D^3 = 35{\cdot}3\,D^3$$

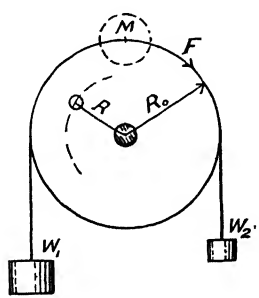
$$\text{and} \quad R^2 = \frac{R_o^2}{2} = \frac{D^2}{8}$$

$$\text{and} \quad wR^2 = I \qquad \text{or} \quad 35{\cdot}3\,D^3 \times \frac{D^2}{8} = 87514$$

$$\therefore D^5 = \frac{87514 \times 8}{35{\cdot}3} = 20{,}000 \text{ nearly}$$

$$\text{And } D = \sqrt[5]{20{,}000} = \underline{7\tfrac{1}{4} \text{ feet.}}$$

Pp. 478 and 680. **Moment of Inertia by Experiment.** —The method to be here described is especially suitable for cylindrical objects such as fly-wheels, pullies, lines of shafting, and the like; though also usable in the case of other objects if placed within a ring for the purposes of the experiment. Referring



Experimental determination of Moment of Inertia.

Fig. 972.

to Fig. 972, a strap or cord passes round the rim of the object and supports two experimental weights W_1 and W_2, of which W_1 is always heavier. Imagine the mass M of the body to be concentrated at the rim, and the frictional force F to be similarly treated. Also R_o = radius at rim, and R the radius of gyration. Of course W_1 begins to descend with uniform acceleration f, the force being

$W_1 - W_2 - F$; and the movable mass is $(W_1 + W_2) \div g + M$. And as f = force ÷ mass,

$$f = \frac{W_1 - W_2 - F}{(W_1 + W_2) \div g + M} = \frac{g(W_1 - W_2 - F)}{W_1 + W_2 + Mg}$$

Also $s = \frac{1}{2}ft^2$ and $2s = ft^2$

$$2s = \frac{g(W_1 - W_2 - F)t^2}{W_1 + W_2 + Mg}$$

giving the fall s in feet, during a time of t seconds.

Next, make a second experiment, as before, with the exception that a small weight w is to be taken from W_2 and added to W_1 so that the difference is greater and the acceleration is increased and the new fall s_1 in time t is given by

$$2s_1 = \frac{g(W_1 - W_2 + 2w - F)t^2}{W_1 + W_2 + Mg}$$

Finally we have, by combination, and in absolute units (lbs. and feet),

$$I = WK^2 = MgR^2 = \left(\frac{wt^2}{s_1 - s} - \frac{W_1 + W_2}{g}\right) g R_0^2$$

In this result R, R_0, s_1 and s are reckoned in feet, t in seconds, W_1, W_2 and w in lbs.

Pp. 492 and 863. **Acceleration Curves.** To correlate the scales of velocity, acceleration, and distance, when the velocity curve is drawn on a *distance base*, refer to Fig. 825, p. 863.

Suppose

velocity increases 1 ft. per sec. $v = 1$

while travelling over 1 ft. $d = 1$

actual velocity being 1 ft. per sec. $V = 1$

Also, let

$\frac{1}{x}$ represent velocity scale

$\frac{1}{y}$,, distance scale

$\frac{1}{z}$,, acceleration scale

Then, from the figure

$$x = \frac{v}{s} \cdot \mathrm{V}$$

$$\therefore \frac{1}{z} = \frac{\frac{1}{x} \times \frac{1}{x}}{\frac{1}{y}} = \frac{y}{x^2}$$

or Fraction for acceleration scale $= \dfrac{(\text{vel. scale fraction})^2}{\text{distance scale fraction}}$

P. 502. **Flexible Couplings.**—It often happens that shafts, though made truly in line when first erected, may take small relative movements through lack of rigidity in the supports, thus causing a tendency to breakage if firmly coupled. Deviations that are small and unimportant at low speeds may become very troublesome at high ones, partly because the rate of change of deviation is greatly increased, and partly, also, because the amount of deviation may itself be multiplied by centrifugal force. As examples we have the very potent one of the breakage of propeller shafts of ships, no doubt largely on account of the flexibility of the hull; and the necessity of universal couplings in motor-cars, to prevent breakage due to bending of the frame, as well as to compensate for slight inaccuracies of workmanship that would otherwise cause hot bearings. When engines and dynamos are coupled direct, there is the same danger, for a very small inaccuracy that would not be apparent on a more slow-running shaft would soon cause heating or knocking on account of the high speed.

Of the flexible couplings illustrated in Fig. 973 the first two are intended for marine shafts, and the third for dynamo work. Brotherhood's coupling consists of a large hollow casting D, to which is bolted a thin corrugated steel disc E; and while the casting is bolted to the shaft A, the disc is bolted to the shaft B. The shafts butt together on a spherical surface C, and any relative bending of the two is met by a buckling of the disc E to of course a very slight extent. The faults of this coupling are its large size and the fact that it cannot well support an axial tension. In Alley's coupling the shafts F and G are supplied with solid

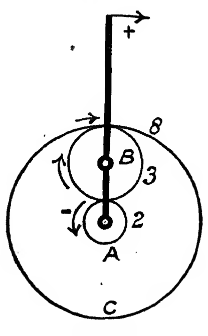
flanges for the bolts. The left flange has a spherical projection K, which fits within two half-discs J fastened to the right flange, and the pivot L bears against the shaft G. Also there are steel bushes in the left flange to receive the barrel-shaped ends of the bolts, which are a loose fit radially. When the axes of the shafts deviate slightly a movement takes place at the surfaces K and L, which movement is freely permitted on account of the clearances at M and N and the sliding of the bolts in the steel bushes. At the same time it must be noticed that either tension or compression in an axial direction is fully resisted, and the whole coupling is not large in diameter. The remaining coupling was used by Dr. Hopkinson for coupling dynamos, and is only adaptable to rather small torques. Its construction is very similar to Brotherhood's, having a hollow casting R keyed to the shaft P, and a disc S keyed to the shaft Q. These are connected by a leather disc T held down to casting and disc by the ring washers U U, and all the metal surfaces are rounded where they meet the bending portion of the leather. This coupling has proved very useful for its purpose.

P. 526. Epicyclic Trains.

Example 78. Two spur wheels, A and B, whose diameters are 2 and 3 respectively, are in gear with an annular wheel C whose diameter is 8. The wheels A and C have a common axis, but B is carried by an arm centred on the axis of A.

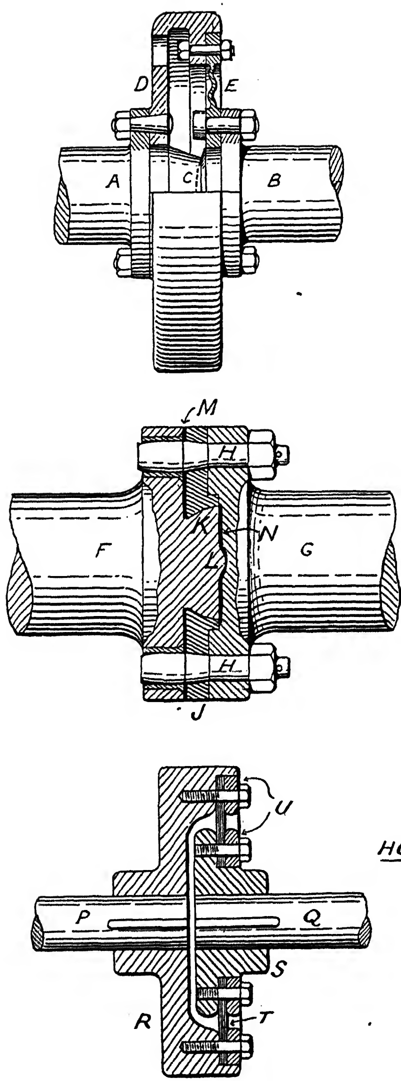
(1). If A makes 5 rotations while C makes one, both in the same direction, find the angle described by the arm during the time.

(2). Let C make 3 rotations in the opposite direction to the 5



Epicyclic Problem.

Fig. 974.



BROTHERHOOD'S

ALLEY'S

HOPKINSON'S

Flexible Couplings. Fig. 973.

rotations of A. Find the rotations and direction of motion of the arm to cause them. (Hons. Applied Mechs. Exam.)

Referring to Fig. 974 and to p. 522.

If A be fixed and the arm rotates +

$$\text{C's turns} = 1 + \frac{A}{C} \quad \text{for one rotation of arm}$$

$$\text{and C's turns} = x\left(1 + \frac{A}{C}\right) \quad \text{for } x \text{ rotations of arm.}$$

While these take place, imagine A to make y rotations, either + or −, *let us say minus.*

$$\text{Total turns of C} = x\left(1 + \frac{A}{C}\right) + y\,\frac{A}{C}$$

Then by Case (1)

$$-1 = x + \tfrac{1}{4}x + \frac{5}{4}$$

$$\therefore\ x = -1{\cdot}8$$

And the arm must turn through $1{\cdot}8 \times 360 = \underline{650^\circ}$ in the same direction as the rotations of A and C.

Again, *By Case* (2): taking A's turns minus,

$$3 = x + \tfrac{1}{4}x + \frac{5}{4}$$

$$\therefore\ x = 1{\cdot}4$$

or the arm must turn through $1{\cdot}4 \times 360 = \underline{504^\circ}$ in the same direction as C's rotations, but opposed to those of A.

P. 530. Lapping of Belts.

Example 79. A rope has its direction changed through two right angles by passing round a grooved guide pulley whose diameter is 12 ins.; the diameter of the axle being $1\frac{1}{4}$ ins., and μ for axle = ·07. How is the efficiency of the pulley affected by axle friction when a load of 2500 lbs. is being raised? If the pulley were fixed so that it could not turn, how would the efficiency be affected by the friction of the rope on the pulley when μ = ·6? (Hons. Applied Mechs. Exam.)

Case 1. $\quad P = 2500$

$$P_1 = 2500 + \left\{(P_1 + 2500) \times {\cdot}07 \times \frac{1{\cdot}25}{12}\right\} = 2536$$

$$\therefore\ \text{Efficiency} = \frac{2500}{2536} = \underline{{\cdot}98}$$

se 2. $$\text{Log}\,\frac{T_n}{t_n} = \cdot 434\,\theta\,\mu = \cdot 434 \times 3\cdot 14 \times \cdot 6 = \cdot 817$$

$$\frac{T_n}{t_n} = 6\cdot 56 \qquad \text{and } T_n = 2500 \times 6\cdot 56 = 16300$$

$$\therefore \text{ Efficiency} = \frac{2500}{16300} = \underline{\cdot 152}$$

ıe comparison is $\frac{\cdot 152}{\cdot 98}$ or only $\frac{1}{6}$ of the efficiency of Case 1.

P. 545. **Pitch-chain Gearing.**—The Brampton chain, g. 975, is much used in motor-car drives. Its construction is ɔt dissimilar to types already described, and is only different in oportion. The links A A are rather thin, but made of tough

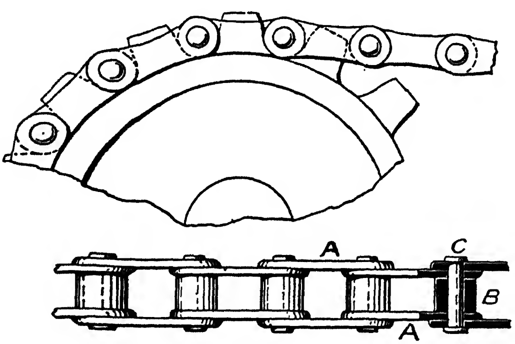


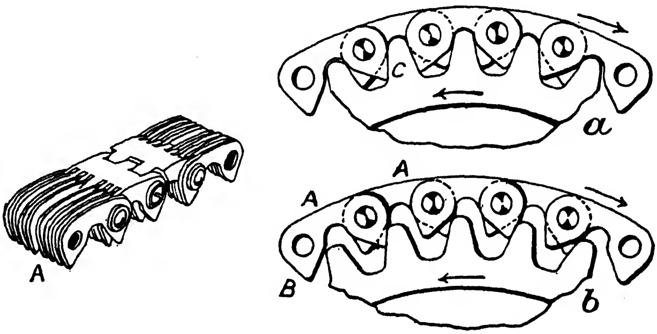
Fig. 975.

The Brampton Chain.

teel, and the inner edges are rounded to facilitate entrance upon he teeth. The rollers B, having to stand wear, are of strong teel hardened, and are as large as possible so as to give a real ınti-frictional advantage. The rivet C is well flattened over by ɔressure.

The Renold chain, Fig. 976, is a very superior chain, nechanically speaking. It consists of a series of very thin plate inks A A, this construction being for the purpose of equally dis- :ributing the stress on the pins, and each link is supplied with

two wedge-like projections B. The pull of the chain causes these to lie firmly in the tooth space, as at C, when new. After wear, the chain rides up the teeth in the manner shewn at *b* until it finds its bearing, but it will be noticed that the meeting surface of link and tooth has a constant angle to the line of pull. The lower view indicates the appearance after the chain has stretched or



The Renold Pitch-chain.
Fig. 976.

worn ·6 in. per foot, before which it would have been adjusted. On account of its perfect action the chain has been used in the Wolseley motor-car with great satisfaction for the very rapid drive direct from the engine, its speed reaching 800 to 1200 ft. per min. The construction of the chain requires one of the wheels to be shrouded both sides so as to hold it in place, and for this purpose the smaller wheel is chosen as a matter of cheapness.

P. 550. **The Dynamo.**

Elementary Principles.—A 'conductor' is any material that will conduct electricity. Applying the term to dynamo work, it may be viewed as a rod or wire of copper arranged to cut or pass across the lines of force of a magnetic field.

The law of magneto-electric induction, discovered by Faraday, affirms that whenever a conductor is moved across such force lines, an E.M.F. (electro-motive force) is set up in the conductor, and a current flow is caused by such E.M.F. if the circuit of the conductor be closed. Also the direction of the current will bear relation to the direction of motion of the conductor, and the intensity of the E.M.F. will depend upon the speed with which the conductor cuts the lines of force.

Any magnet, permanent or electric, causes a state of electric stress in the medium between its poles (Fig. 977), and the intensity of the stress is conveniently measured as 'force lines per sq. centimetre' on the cross-sectional area of the poles. If N = total number of force lines, and A_t = ampere turns of the coils of an electro-magnet (viz., Amperes sent through × Turns of the Coil):

$$N \propto A_t$$

in any particular magnetic circuit.

Consider now the application of Faraday's law in a simple manner, Fig. 978.. Two rods D D are connected by a galvanometer C and a loose rod *e*, thus forming a closed circuit, the whole being placed across the force lines existing between the magnet poles A and B. If *e* be moved to right or left to cut these lines, an E.M.F. is set up in the circuit, which is detected by the deflection of the galvanometer needle. The direction of current thus caused depends on the direction of motion of *e*, and the E.M.F. on its speed.

Next imagine a rotary-moving circuit, Fig. 979, consisting of a stiff wire A connected at its open ends to the cylinders B B, and forming a *pair* of conductors a_1 and a_2. The circuit is actually completed by brushes from slip rings connected to the line wire C. Looking at the end view, this '**Armature**' is placed between the poles P P of the electro-magnet E M, and is there rotated uniformly. When the wire is in the position E, the movement of its outer edge is *along* the force lines, no cutting is done, and no E.M.F. is produced; but when at D the lines are cut at the highest speed, and the maximum E.M.F. exists in the conductor. A careful examination will shew that the direction of E.M.F. is changed twice in a revolution, and the diagram is

therefore a harmonic curve F. Also the two conductors, being in series, will produce twice the voltage of one, and each will cut the lines twice in a revolution. If n = revs. per sec. and c = number of conductors round the circle of rotation,

$$\text{Average E.M.F.} + \text{or} - = \frac{2nNc}{10^8} \text{ volts.}$$

This apparatus constitutes an elementary Alternating-current Dynamo or **Alternator.**

If the conducting cylinder, B, be split as in Fig. 980, the direction of the current is re-changed at each half-revolution, making the E.M.F. always of one sign, and the cylinder is then known as a **Commutator**, the machine being the elementary form of the Direct- or **Continuous-current Dynamo.** Also the number of splits will correspond with the number of conductors or conductor coils. The diagram of E.M.F. will be as shewn at G, if for a pair of conductors, and the more splits there are, the more uniform will be the voltage, as at J, giving a steady continuous current. In actual dynamos, through necessary motives of construction, the current is divided into halves when passing through the armature conductors, and the E.M.F. is therefore also halved.

$$\text{Average E.M.F.} = \frac{nNc}{10^8} \text{ volts.}$$

When the armature is rotated, and a current produced, the machine is called a **Generator**, but if a current sent through an armature causes rotation, a **Motor** is constituted. The principle is identical in both cases, for a motor is only a reversed generator, with some small differences in detail.

Magnet (or Field) windings. There are only two principal magnet forms in modern work, the horse-shoe K, Fig. 981, and the multipolar magnet L, the latter being most likely to persist. The field may be 'excited' (1) by a separate dynamo W at M, common in alternator work, or (2) by current produced from the dynamo itself, as in direct-current (D.C.) machines. The latter method is practised in three different ways: N, where the line wire passes continuously through the armature and round the magnet in series, forming a **series-wound** dynamo: R, by

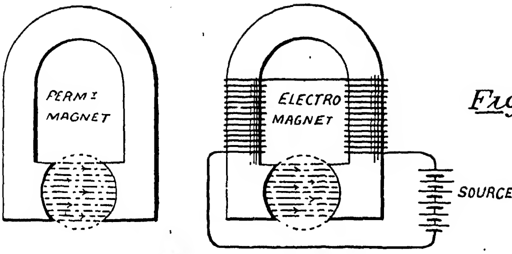


Fig. 977.

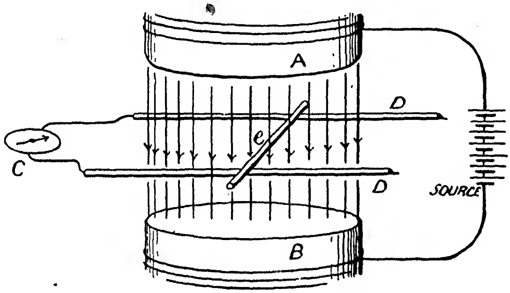


Fig. 978.

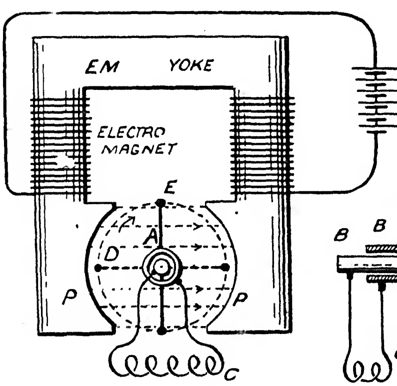
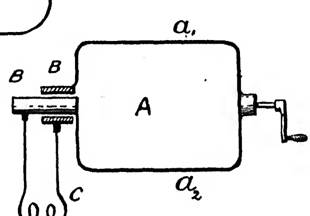
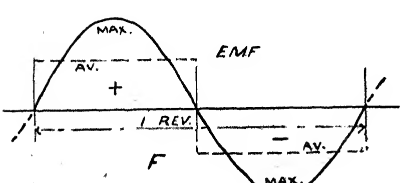
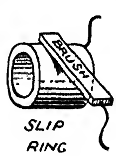


Fig. 979.







The Dynamo.

current taken from the brush terminals through a shunt wire round the magnet, termed **shunt-wound**: and Q, where two windings are laid round the magnet, one from the circuit, and one a shunt, making a **compound-wound** dynamo.

In the case of Alternators it has become the fashion to call the moving part the **Rotor**, and the stánding part the **Stator**; so that either armature or magnets may be revolved under this nomenclature.

Transformers. A *direct* current cannot be easily transformed, that is, have the relation $\frac{E}{C}$ altered in value while $E \times C$ remains approximately constant, and $E \propto \frac{1}{C}$ The alternating current can be easily so treated. Shewn simply in Fig. 982, we have the principles of the induction coil, where on the primary side a current of low E and high C is transformed into a high E and low C in the secondary. The reverse operation is also possible, so that there are both *step-up* and *step-down* transformers.

Rotary Converters are a means of changing an alternating current into a direct current, and *vice versa*, consisting of a single dynamo, Fig. 983, having a split commutator at one end of the armature for D. C., and slip rings at the other end for A. C. Only one definite relation is possible between the two sides, as for example from 400 or 500 volts D. C. to 300 volts A. C. in a three-phase machine. If further alteration be required, a static transformer T must be put in the alternating circuit. This machine is remarkably adaptable, however: it may be used (1) to change existing current to or from either form, (2) to change part of the current and deliver the rest mechanically to the machine as motor, (3) as a generator supplying either form of current, or both.

Motor Generators consist of at least one pair of machines, one of which is a generator and the other a motor. In Fig. 984 the motor M receives an alternating current, and drives the dynamo D, from which a direct current is evolved. Any desired alteration in the ratio $\frac{E}{C}$ can be procured when first designing.

Motor Dynamos.—These are true transformers of direct urrent, and consist of motor and dynamo on one bed. The

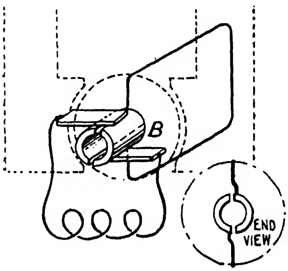
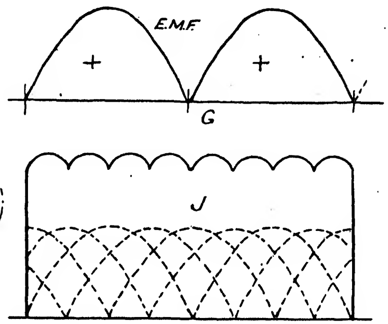


Fig. 980.



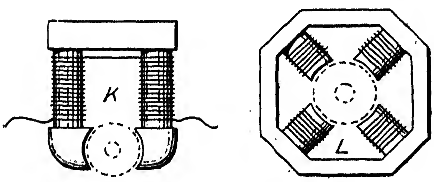
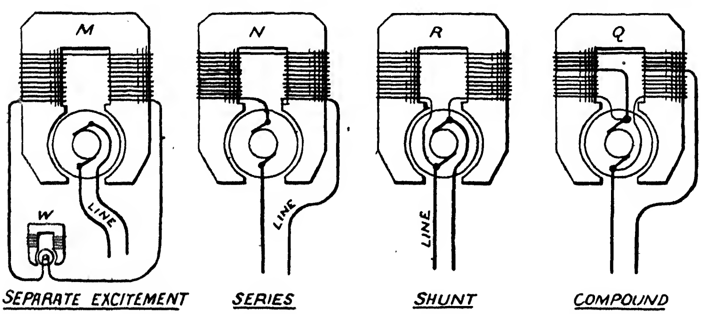


Fig. 981.



The Dynamo.

current E C, Fig. 985, drives the motor M, which is coupled to a generator D producing a current E C, and the transformation may be whatever is required for the particular purpose.

Boosters are motor-dynamos applied to automatically increase or 'boost' the voltage of a line already supplied with current. There are two principal uses for boosters: (1) when a storage battery requires high voltage for charging during light load at a station, and a booster is put in series between the main generator and the cells to raise the voltage to the required amount, as in the figure; (2) when the inevitable voltage drop in long lines is made up in a similar manner.

Motor Starters.—On starting a motor the current would reach an excessive value if it were turned on to the motor instantly. To prevent this a multiple-step rheostat is used to keep the current to a reasonable value, the surplus being absorbed by resistances that are cut out one by one till full speed be attained. Referring to Fig. 986, there are three terminals A, B, and C on the switch board, connected respectively to the main supply S, the field winding, and the armature. Also B is coupled to the field switch, and C to the armature switch, each of which is supplied with a number of resistances. In the figure the switch arm is shewn on the vulcanite insulators V V, and current is cut off. Moving the arm downward, the field winding is gradually supplied as R_1 and R_2 are cut out, after which the armature resistances R_3 to R_7 are similarly eliminated, the motor speed being finally regulated on the armature circuit.

Pp. 553 and 961. **Electric Accumulators.**—Secondary cells have improved considerably. Their efficiency is usually reckoned on the combined charge and discharge. Stated on what happens to the cell itself, without regard to the engine and dynamo:—

$$\text{Current efficiency} = \frac{\text{ampère-hours discharge}}{\text{ampère-hours charge}} = \text{about } 80\%$$

$$\text{Energy efficiency} = \frac{\text{Watt-hours discharge}}{\text{Watt-hours charge}} = \text{about } 60\%$$

The reason for the lower result in the second case is the decrease of voltage per cell from 2·14 to 1·9 volts between charge and discharge. Naturally, however, the second statement is the one having most value for the engineer.

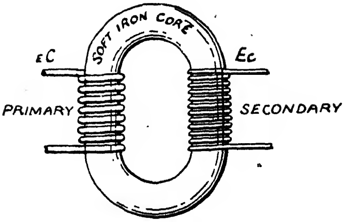


Fig. 982.

TRANSFORMER

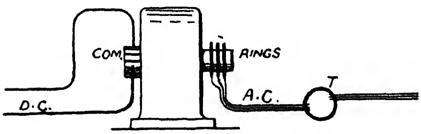


Fig. 983.

ROTARY CONVERTER

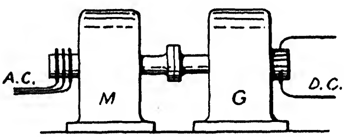


Fig. 984.

MOTOR GENERATOR

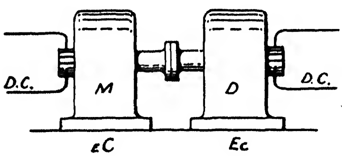
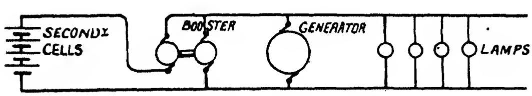


Fig. 985.

MOTOR DYNAMO



BOOSTER TO SECONDARY CELLS

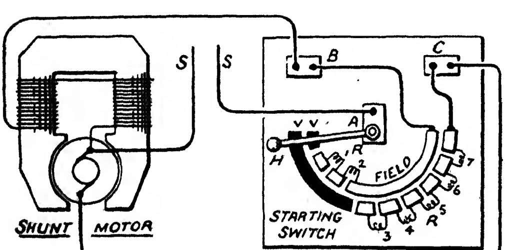


Fig. 986.

Dynamos and Motors.

P. 568. **Ball Bearings.**—Fig. 987 shews a recent form of ball-bearing cage having some interesting features. The balls are made of hard tool steel, and are sprung between the two rings

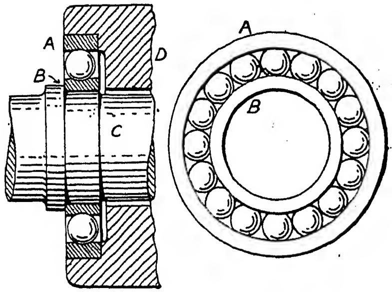


Fig. 987.

Ball Bearings

A and B, also of tempered steel, the latter having concave grooves to receive them. The rings are then forced into place, B upon the shaft C and A within the bearing or wheel D. The system is being applied pretty extensively in motor-car practice.

P. 569. **Friction Clutches.**—The design of a friction clutch will depend very much upon the use to which the clutch is to be put. Thus Fig. 589 is a case where slipping only is to be provided for, and Fig. 588 is meant only for disengaging. Nowadays, as for motor-cars, clutches are required that will not only engage and disengage without shock, but will also slip to any desired extent within the limits of tight and free, and thus supply a very perfect variable speed gear. A combined slipping and disengaging clutch should be free from any tendency to stick, the wear should not be considerable, and the frictional heat should be constantly and automatically removed. The Bagshaw clutch, Fig. 988, is very simple, and is a disengaging clutch only. One shaft A is provided with a drum B, in which is a split ring J running freely within it when out of gear. The other shaft C carries a disc D, through which passes the wedge F. When the wedge is moved leftward by the lever boss E it separates the two levers G G, that in turn expand the ring J at the point H, and thus cause the gripping connection between the shafts. The

Dohmen-Leblanc coupling, Fig. 989, is also for gripping only. The drum B is carried on the shaft A, and the disc C on the shaft K, while gripping jaws D D with corrugated surfaces slide in guides formed on the disc. These grips are connected to the lever boss E by stout flat springs F F, that force the parts D and B into contact when E is moved leftward. The interesting point in this clutch is that when the boss E is moved hard over, the pin G gets past the vertical through H, and so locks the clutch parts as to prevent any tendency to release when driving.

The remaining diagrams illustrate the newest forms of clutch, suitable both for hard grip and for the most sensitive slipping. The Hele-Shaw clutch, Fig. 990, consists of a solid drum E keyed to the shaft A, and a hollow drum C with sides B and D, the part D being keyed to the other shaft F. The drum E is supplied with teeth *a*, and the drum C with teeth *b*; and of the discs G, half are fixed to one drum, while the alternate half are keyed to the other drum. The discs are provided with circular wedge-form corrugations, each containing an angle of about 35°, and the gripping is obtained as in the Weston clutch by pressing the discs together by means of the plate H. The boss K causes the pressing action through the pins P, but the lever ring M only does this indirectly at first. Moving M leftward it first tilts the small levers L till it falls into the hole D, thus permitting the compressed spring J to expand and press K and P upon plate H. Further pressure can now be obtained by ring M on the boss K. To disengage, the lever moves M rightward, and the levers L fall out of gear by the action of spring N, thus keeping the clutch out of gear. The Coil Clutch, Fig. 991, is both simple and effective. The drum B is fixed to shaft A, and carries a disc C, through which passes the lever boss D. A drum F is keyed to the other shaft E, and a coiled spring G lies loosely upon it, being anchored by the eye H to pin L in the drum. A lever M, centred on the pin N in the disc C, is moved radially when acted on by the boss D, and then engages the lug J on the other end of the coil, causing it to grip upon the drum F. The grip of such a coil, as is well-known, is considerable, even though the tension may be slight, and there is no tendency to release in the design shewn because the lever lies in the parallel portion of D when in full action. This clutch will

slip without trouble if kept lubricated, indeed both the clutches Figs. 990 and 991, are intended to be filled with oil under circulation.

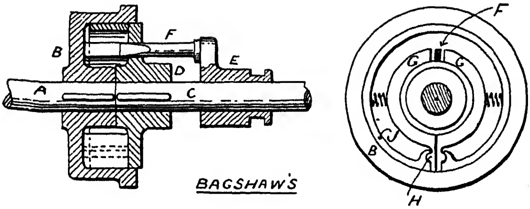
Pp. 577 and 872. **Efficiency of Worm Gear.**—Extensive experiments were made by Messrs. Wm. Sellers & Co. to determine the efficiencies of worm and spur gear at various speeds, and the results are set out in Fig. 992. The upper curve is for simple spur gear, where the shafts are of course parallel, or enclose an angle of 0°; and the angle w, enclosed by the tooth line and a normal to the shaft, is 90°. Here the efficiencies rise from 90% to 98% as the pitch-line speed is varied from 3 to 200 ft. per m. The remaining curves shew the efficiencies of worm gear when designed as in the diagram. Taking the lowest curve the angle of thread is 5°, and the shaft angle is 85°, causing the efficiency to vary between 40% and 78%. These figures include all bearing losses.

The frictional resistances in worm gear are caused by (1) the wheel and worm bearings, (2) the roll of the teeth, (3) the sliding of the thread, (4) the thrust of the worm. The last two are the greatest of all, and every effort should be directed towards their decrease. Mr. C. W. Hill plotted the curves in Fig. 993 from the theoretical formula:

$$\text{Efficiency} = \frac{1 - \mu \dfrac{p}{2\pi r}}{1 + \mu \dfrac{p}{2\pi r}}$$

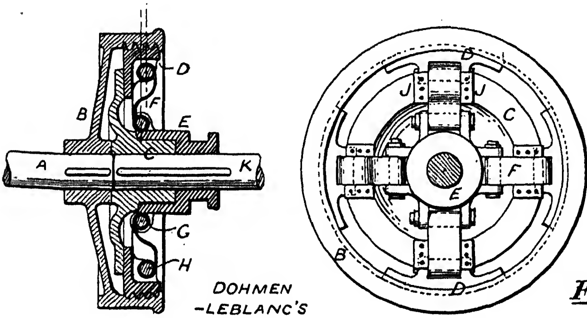
Where μ = coefficient of friction.
p = pitch of teeth or of thread.
r = pitch radius of driver, worm, or wheel.

The left-hand set of curves are when the worm drives the wheel, and the right-hand set when the wheel drives the worm: and each curve is for a different value of μ. The efficiency thus stated, it must be noted, refers only to the sliding friction of the teeth, and all the before-mentioned losses must be included if the worm gear is to be considered as a whole. The diagram shews some interesting facts, however. Thus, the maximum efficiency is reached when the thread angle is about 25° to 30°, which



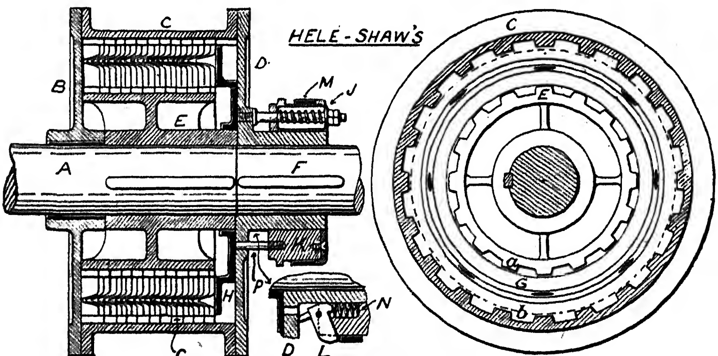
BAGSHAW'S

Fig. 988.



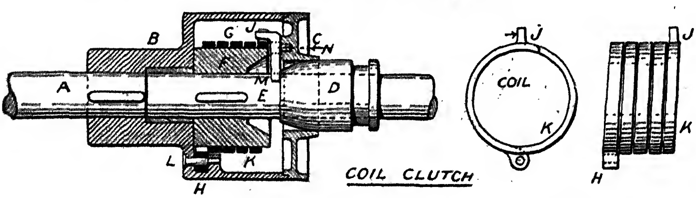
DOHMEN
-LEBLANC'S

Fig. 989.



HELE-SHAW'S

Fig. 990.



COIL CLUTCH

Fig. 991.

Friction Clutches.

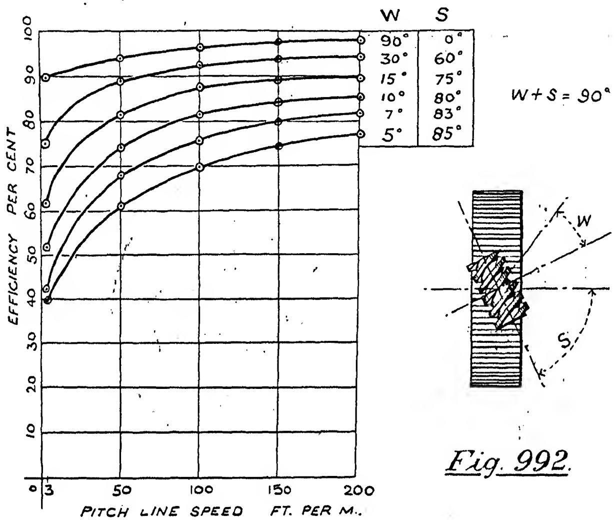


Fig. 992.

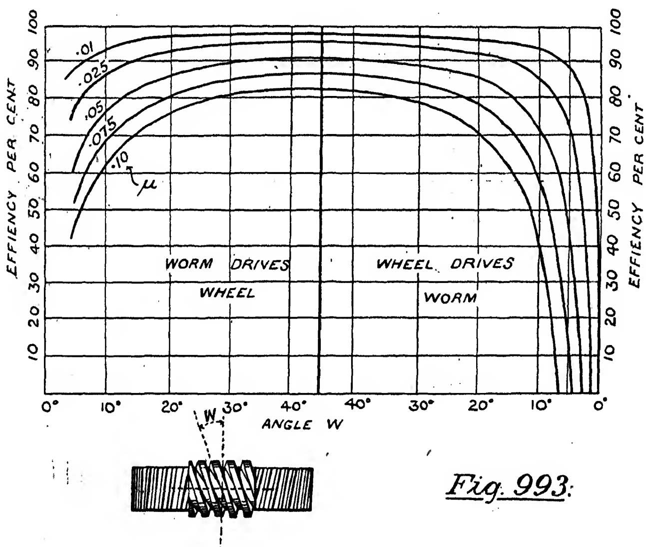


Fig. 993.

Worm-Gear Efficiency.

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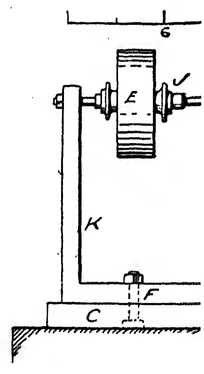

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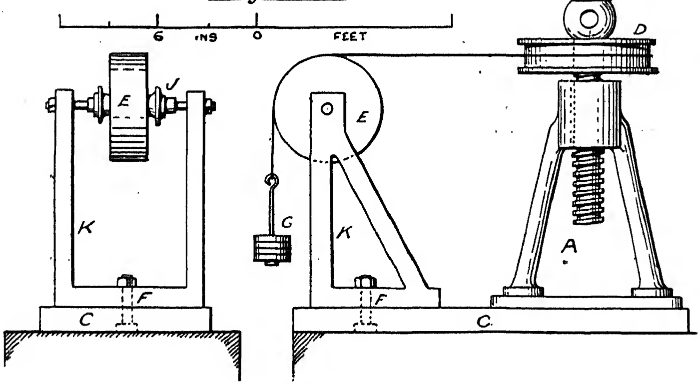
means a worm gear that will overhaul. If the worm is to hold the wheel we refer to the lower end of the right-hand curves. The points where these intersect the base line indicate the angles of friction, and consequently the angle of thread for a retaining worm.

P. 874. **Efficiency of Screw.**—The apparatus required for the experiment described on p. 874 is represented in Fig. 994. A common 2-ton screw-jack A is mounted on a baseboard C, being further supplied with a pulley D, a bolt B to hold the weights, and a plate L to support them: the last-named being

Apparatus for Screw Experiment.

Fig. 994.



securely fastened to the screw-head. A string from D passes over a pulley E, running on bicycle ball-bearings J, and carries the small weights G that are equivalent to the effort. The bracket K is held by one central bolt F so that it may be swivelled in plan to allow the cord to lead off tangentially to D for the two experiments of raising and lowering. The friction of the ball bearing is inappreciable, and can be neglected altogether, so that if G be

delicately adjusted to descend uniformly against a load H, when once set in motion, the value of G is the P_1 required as at p. 956 and the usual set of curves may be constructed. The load being taken as H plus weight of screw, the $P_1 : W$ curve passes through the origin.

CHAPTER X.

Pp. 587 and 879. **The Platinum Resistance Thermometer.**—It was Sir William Siemens who first indicated temperature by the measurement of electrical resistance, constructing his resistance thermometer in 1871. Some difficulty in finding a material that would not change with time and temperature led to the instrument being practically abandoned until its improvement by Callendar and Griffiths, who adopted a platinum wire shielded so as to resist the corrosive action of certain gases. The resistance-measuring apparatus has also been much improved.

Specific Resistance. This term means the electrical resistance of any particular material, and is defined as the *resistance in ohms of a wire one centimetre long and one square centimetre in cross-section, the temperature being 32° F.* Electrical resistance increases with increase of temperature,

$$R_t = R_0(1 + a\,t^{\circ})$$

where a is the temperature coefficient. Strangely enough this has the same value as the coefficient of gas expansion (p. 589), so the zero of resistance is coincident with the absolute zero of temperature, and resistance vanishes at that point. These results are shewn in Fig. 995, where specific resistance and temperature have been correlated. Exact measurement of resistance will therefore serve as an accurate estimate of temperature.

Measurement of Electrical Resistance. Fig. 996 is 'Wheatstone's network' of conductors. A wire A H K D, of diamond shape, is connected to a galvanometer between H and D, while its ends A K are similarly linked to a battery B. Let the wires be adjusted in their lengths until the galvanometer reads zero, when no current will be passing between H and D. The whole flow is

therefore passing from B to A, then dividing along the arms A H K and A D K, finally returning by K B. By Ohm's law, $E = C R$; and when E is constant C R is constant also. This is the case at A, for the current is undivided, and must also be the case at H and D, though with a lower value of E, for none of the current is lost. Therefore

$$C_1 R_1 = C_2 R_3$$
$$C_1 R_2 = C_2 R_4$$

Dividing the first equation by the second,

$$\frac{R_1}{R_2} = \frac{R_3}{R_4}$$

The practical use of the network is to measure one unknown resistance when the others are known, and in this form becomes the Wheatstone Bridge, Fig. 997, where the same letters have been retained to facilitate understanding. Upon a wood base is fixed a metre scale *lm*, some brass plates *a* A *n*, *pq*, *r* K *k*, and a 'slide-wire' *ak*. A standard known resistance is placed at R_2, joining *qr*, and the unknown resistance R_1 is between *n* and *p*. The button D is then moved along the wire till, when depressed, there is no movement of the galvanometer. Then the resistances R_3 and R_4 are represented by the lengths of metre scale intercepted, and the unknown resistance is

$$R_1 = \frac{R_3 R_2}{R_4}$$

The Thermometer. Fig. 998 shews the present form of resistance thermometer. A long loop of fine platinum wire K is coiled round a piece of mica of cross-shaped section, and the two ends of the loop are welded to the thicker platinum leads B emerging at the terminals C D. The object being to measure the variation in resistance of the coil K only, a separate loop E, equal in every way to the leads B, is so connected through F and G to the measuring apparatus that the final reading is the difference of E's resistance from that of B and K together; and the net result is of course the resistance of coil K alone. Fig. 999 is the measuring bridge. The thermometer coil and temperature leads *bb*, together with the wire from A to D, represent the resistance R_1: the compensating leads *ee*, the wire E D, and adjusting coil *r*, constitute R_2:

while the remaining resistances R_3 and R_4 are placed between A H and H K respectively. The battery B is connected to A K, and the galvanometer to H D. The thermometer being set to freezing point, the pointer D is rotated till at D, and the resistance r is adjusted till the galvanometer G reads zero. D_1 then indicates 32° F, or 0° C, according to the scale adopted. If N be now subjected to a higher temperature, G will be out of balance till D be sufficiently raised, and the point on the circle where balance is restored will show true temperature of N, for the leads *ee* and *bb* are mutually opposing resistances, and the net result will be the resistance of coil N only. The platinum thermometer gives consistent readings up to about 1000° C. or 1800° F. It is also very delicate, and very respondent, the changes being taken up without any measurable lag, and it is equally true for high or low temperatures. By balancing N's resistance against one slightly less, a dead-beat galvanometer will autographically trace a temperature-time curve (see p. 876).

P. 590. **Combination of Boyle's and Charles' Laws.**—The formula $PV \propto \tau$ can be rigidly proved by step-by-step process. In the following columns the pressure, volume, and absolute temperature of a gas are gradually changed, the first three lines under Boyle's law, where pressure × volume is constant and temperature invariable; and the last two lines under Charles' law, where volume ÷ absolute temperature is constant, and pressure invariable. In either case multiply or divide out the respective columns and verify.

	Pressure.	Volume.	Abs. Temp.
By Boyle.	P	V	τ
	1	P V	τ
	P_1	$\frac{PV}{P_1}$	τ
By Charles.	P_1	$\frac{\frac{PV}{P_1}}{\tau}$	1
	P_1	$\frac{\frac{PV}{\tau_1 P_1}}{\tau}$	τ_1

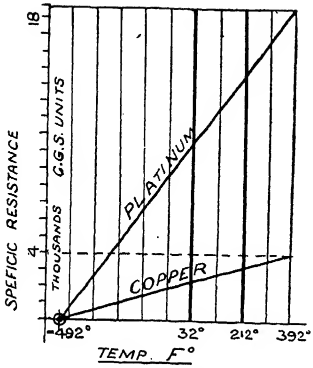


Fig. 995.

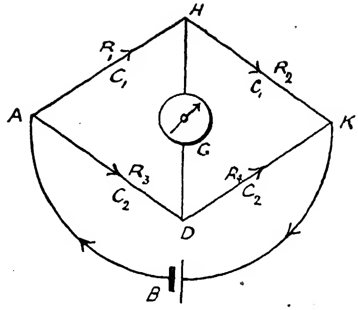


Fig. 996.

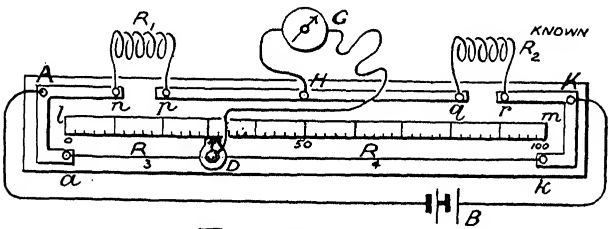


Fig. 997.

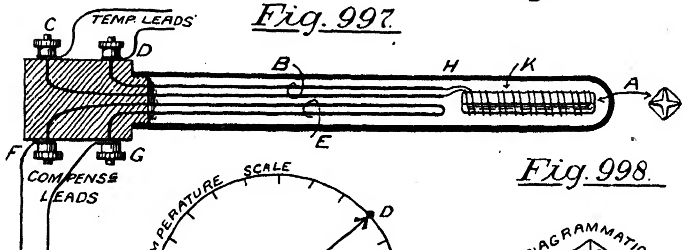


Fig. 998.

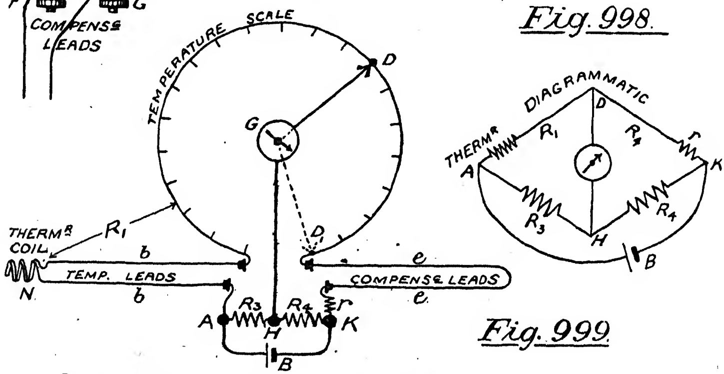


Fig. 999.

The Platinum Resistance Thermometer

Hence the volume at temperature τ_1 is :—

$$V_1 = \frac{P V \tau_1}{P_1 \tau}$$

$$\text{or } \frac{P_1 V_1}{\tau_1} = \frac{P V}{\tau}$$

$$\therefore \frac{P V}{\tau} = \text{constant}, \quad \text{and} \quad P V \propto \tau$$

Pp. 600 and 930. **The Mechanical Equivalent of Heat.**—The apparatus of Prof. H. A. Rowland of Baltimore, for the determination of this important value, and already mentioned at p. 930, is illustrated in Fig. 1000. The paddles are rotated by

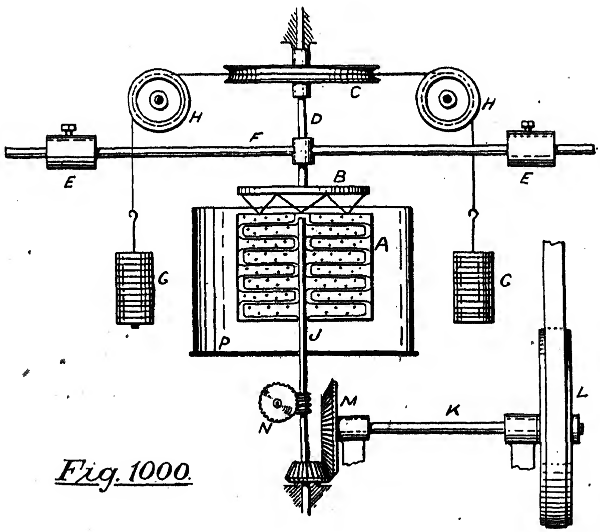


Fig. 1000.

The Mechanical Equivalent of Heat.
Rowland's Apparatus.

power, through pulley L, shaft K, and multiplying gear M to the shaft J. The fixed wings are held stationary by the box A, which is connected to the shaft D by the plate B, and the torque is measured by the weights G G, which are connected to opposite

sides of the pulley (or brake drum) C, over the guide pullies H H. The method is similar to that adopted in engines to measure the brake horse-power, and to avoid vibration and jerk, a long arm F, having weights E E upon it, is fixed to the shaft D, thus absorbing the smaller changes of torque. Lastly, a worm wheel N, serves as a counter, and the mechanical energy produced in foot-pounds is $2\,G \times 2\,\pi\,R\,N$, where R is the radius of C in feet and N the revs. per min. Also the heat energy will be the weight of water × rise of temperature, and the first result will be J times the second. The box P is to shield A from air currents. Rowland advanced the equivalent from 772 to 774 by this apparatus, and still further to 778 by improving his thermometers over those used by Joule.

P. 605. **Expansion Curve Areas: proofs.**

(1) $PV^n = C$

Let a gas expand in such a way that its pressure-volume curve can be represented by the curve $PV^n = C$.

Taking Fig. 612 and remembering that:

$$PV^n = P_1V_1^{\ n} \quad \ldots\ldots\ldots\ldots \quad (a)$$

$$P_1V_1^{\ n} = P_2V_2^{\ n} \quad \ldots\ldots\ldots\ldots \quad (b)$$

$$P = \frac{P_1V_1^{\ n}}{V^n} \quad \ldots\ldots\ldots\ldots \quad (c)$$

We have:

$$\text{Area} = \int_{v_1}^{v_2} P \cdot dV$$

(By *c*)

$$= P_1V_1^{\ n} \int_{v_1}^{v_2} \frac{1}{V^n} \cdot dV$$

Integrating

$$= P_1V_1^{\ n} \cdot \left(\frac{V_2^{\ 1-n} - V_1^{\ 1-n}}{1-n}\right)$$

(By *b*)

$$= \frac{P_2V_2 - P_1V_1}{n-1}$$

(2) $PV = C$

It is required to find the area under a rectangular hyperbola, where $P \times V$ is constant.

Now $PV = P_1V_1$ and $P = \frac{P_1V_1}{V}$ ………… (*d*)

$$\begin{aligned}
\text{Area} &= \int_{v_1}^{v_2} \mathrm{P}\,.\,d\mathrm{V} \\
&= \mathrm{P}_1\mathrm{V}_1 \int_{v_1}^{v_2} \frac{1}{\mathrm{V}}\,.\,d\mathrm{V} \\
&= \mathrm{P}_1\mathrm{V}_1\,(\log_e \mathrm{V}_2 - \log_e \mathrm{V}_1) \\
&= \mathrm{P}_1\mathrm{V}_1\ \log_e \frac{\mathrm{V}_2}{\mathrm{V}_1} \\
&= \mathrm{P}_1\mathrm{V}_1\ \log_e r
\end{aligned}$$

P. 612. **The Reversed Heat Engine.**—Let Fig. 1001 represent the two entropy-temperature diagrams of a perfect engine (1) when taking in heat at τ_1 and rejecting at τ_2 direct working; and (2) when taking in heat at τ_1 and rejecting at τ_1 reversed working. The heat quantities are shewn by areas; or by temperatures only, since the entropy bases are equal. Let $\mathrm{B} = \frac{9}{10}\,\mathrm{A}$. Taking efficiency to mean 'energy utilised ÷ energy received,' we have:

$$\left.\begin{matrix}\text{Working}\\ \text{Direct}\end{matrix}\right\}\ \text{Efficiency} = \frac{\mathrm{C}}{\mathrm{A}} = \frac{\tau_1 - \tau_2}{\tau_1} = \frac{1}{10}\ \text{in this case.}$$

$$\left.\begin{matrix}\text{Working}\\ \text{Reversed}\end{matrix}\right\}\ \text{Efficiency} = \frac{\mathrm{A}}{\mathrm{C}} = \frac{\tau_1}{\tau_1 - \tau_2} = \frac{10}{1}\ \text{in this case.}$$

The result for direct working will not be questioned, for A is the heat received from the boiler, and C is that utilised in the cylinder. The case for reversed working needs a little thought. Here the work done by the piston on the gas is the 'energy received' and is C, while the useful heat produced is two-fold: the change of that work into heat, and the automatic supply from the cold body represented by B. The total heat in the outgoing gas is therefore C + B = A.

Viewing the two engines quite separately, we see then, taking the supposed figures, that when working direct we only utilise *one-tenth* of the energy received; but on reversed working as a *heat supplier*, we return *ten times* the energy we expend. Nine of these ten parts have been extracted from the atmosphere, and we have the curious result of an efficiency considerably greater than

100 %. It is clear, however, that if the two engines are considered as one, the energy B that is rejected on direct working is simply returned on reversal, the work given up at C being used to effect that reversal.

When the reversed engine or heat pump is applied to refrigeration, it is as a *heat extractor* and not as a heat supplier that it becomes useful. Then, adopting a new term instead of 'efficiency,'

$$\left.\begin{array}{c}\text{Co-efficient of}\\ \text{Performance}\end{array}\right\} = \frac{\text{Heat extracted}}{\text{work expended}} = \frac{B}{C} = \frac{\tau_2}{\tau_1 - \tau_2} = \frac{9}{1} \text{ say.}$$

In the two diagrams, Fig. 1002, A is a radiator and A_1 an absorber of heat to or from the outside air, while B B are expanding and C C compressing pistons only, worked from the engines E E. The valves are so arranged as to cause the enclosed air to move in the direction of the arrows.

In the left diagram the air is compressed by C to a high temperature and passed on to the radiator A, where the *heat is given to the outer air.* It is next expanded in B, doing work on the crank, and, falling in temperature, enters the chamber in the *cold condition.*

In the right diagram the air is expanded in B below the outside temperature, and passing through A_1 *absorbs heat from the outer air.* It is next compressed in C to a higher temperature, and enters the chamber in the *hot condition.* The first is a means of cooling, and the second of heating the air of a chamber or room, and,

$$\text{For (1) ....} \quad \frac{\text{Heat extracted}}{\text{work done}} = \frac{\tau_2}{\tau_1 - \tau_2}$$

$$\text{For (2) ....} \quad \frac{\text{Heat delivered}}{\text{work done}} = \frac{\tau_1}{\tau_1 - \tau_2}$$

This method of heating, suggested by Kelvin in 1852, by which the coal, instead of being burnt in a fireplace, supplies a boiler, direct engine, and heat-pump in series, can be made to return more energy from the outer air than is rejected thereto, as follows:

Referring again to Fig. 1001, it has already been shewn that if the direct and reversed engines are coupled, while receiving and delivering heat through the same temperature ranges, the received

and delivered heat are exactly equal. But if the direct drop be large, and the reversed rise of temperature small, we shall have increased the entropy base, and an overplus of heat will have been obtained from the cold body. This is distinctly shewn by Fig. 1003, where the work areas C and E are naturally equal, while only the heat H is rejected to the cold body, the larger heat G is absorbed on reversal, and though the heat D was first supplied by the coal, the larger quantity F is returned at a lower temperature τ_3 viz., through a decreased rise of temperature $\tau_3 - \tau_2$. It must be understood that the direct and reversed operations are here performed in different machines.

Pp. 614 and 884. **Initial Condensation and Leakage.** The first report of the Steam Engine Research Committee of the Institution of Mechanical Engineers, was made by Professor Capper, in March 1905, and detailed at considerable length the trials made upon a horizontal engine, both with and without jackets, condensing and non-condensing. The leakage of the slide-valves was measured by blocking the steam ports, and allowing the lost steam to escape by the exhaust. Leakage was decreased by lubrication. The leakage of the piston was measured by blocking one steam port and running the engine single-acting with the other.

The general conclusions for the particular engine under experiment were (1) that leakage was nearly independent of sliding speed and proportional to pressure difference, inversely proportional to lap, and amounted in quantity to over 20% of the steam supply: (2) that the initial condensation, taken separately from leakage, *diminishes* with increase of initial temperature: (3) that the total missing quantity *increases* with temperature increase: (4) that re-evaporation is often greater without than with jackets. Lastly by plotting heat units for I.H.P., to a base of initial pressure in both the jacketed and non-jacketed trials, the exact point was found for temperature and speed when the jackets could be dispensed with.

P. 616. **The Steam Engine Indicator.** — The usual forms of indicator are unsuited to engines of extremely high speed such as are now adopted on motor-cars. The objection of the

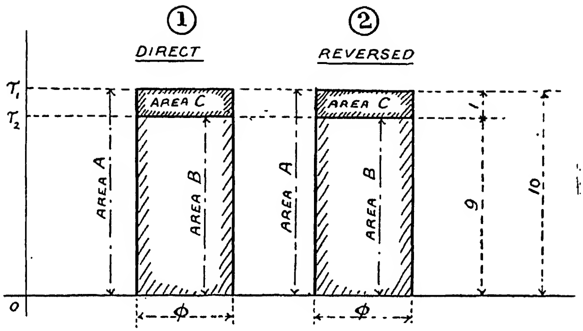


Fig. 1001.

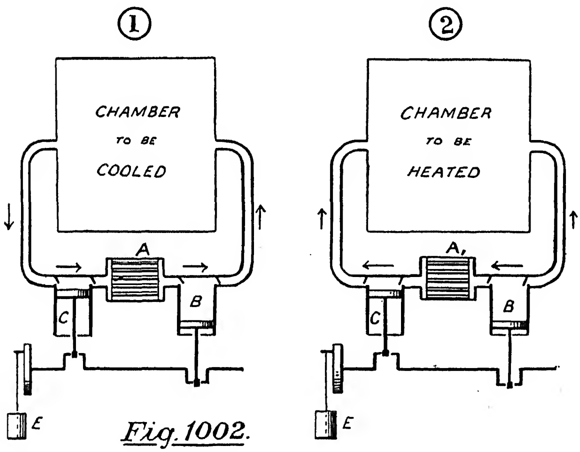


Fig. 1002.

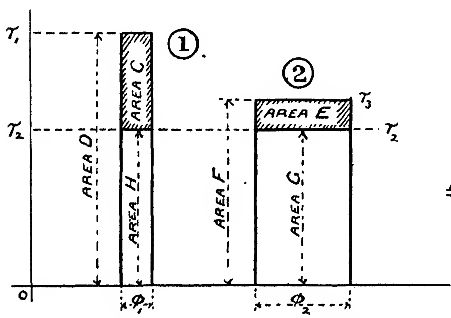


Fig. 1003.

The Reversed Heat Engine.

inertia of the indicator piston and enlarging levers has, however, been overcome in the Carpentier-Hospitalier Manograph, in

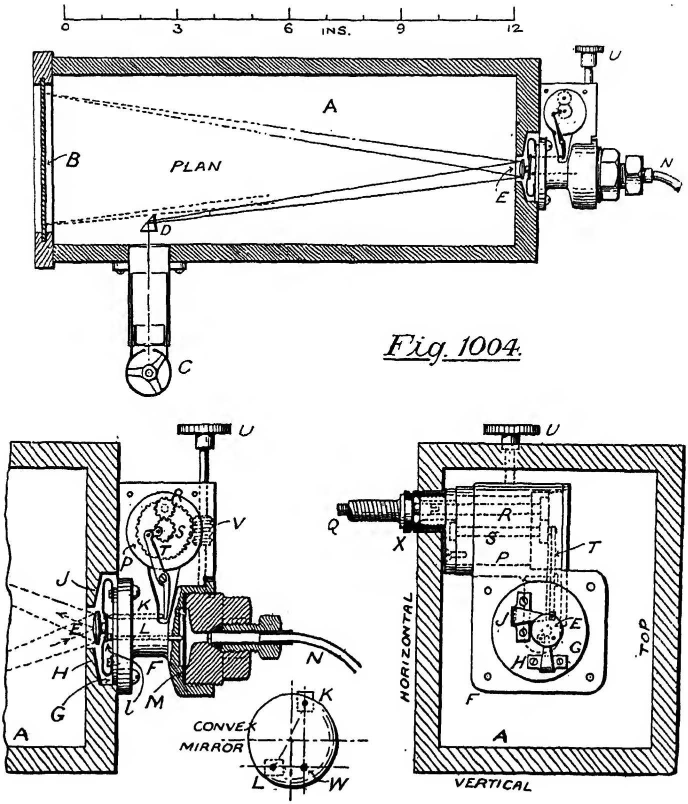


Fig. 1004.

The Carpentier-Hospitalier Manograph.

OR ENGINE INDICATOR BY LIGHT.

which the necessary movements have very small amplitude, while the enlargement is entirely effected by a beam of light. Referring to Fig. 1004, A is a box or light-tight camera, having a ground glass or photographic plate at B, and indicating mechanism at the

opposite end. The important part of this mechanism is the convex mirror E, which receives a ray of light from the acetylene lamp C after reflection by prism D, and projects it upon B. The larger drawings shew the mechanism more clearly. A gun-metal casting F fixed to the box by the seat G, contains the plungers K and L whose motions represent engine stroke and pressure respectively: the pressure movement being caused through the buckling of a thin diaphragm M by the engine pressure in a small-bore pipe N. The plunger K is actuated by the 'reproducer' Q R S T, so called because it imitates the engine stroke, and consisting of a flexible shaft Q, connected at one end to shaft R, and at the other to the crank shaft of the engine. Two equal spur wheels continue the motion to shaft S, upon which is a crank pin and link to rock the lever T. The proportions of these levers must be such as to copy on the average the motion of the crank and connecting rods of the engines usually indicated; and the bearings of the two shafts are contained in the larger bush P, which may be rotated into any required position round the centre S, by the milled wheel U and worm V. The object of this change is the synchronisation of T's phases with those of the engine stroke, which cannot be done till the flexible shaft has taken its drive. The mirror is supported at three points K, L and W, which together form a right-angled triangle; and the point W is a rigid fulcrum or pivot, but the supports K and L are the springs J and H respectively. The plunger K, pressing behind spring J, moves the mirror in a horizontal plane to represent stroke; while the plunger L, pressing behind spring H, moves the mirror in a vertical plane, or at right angles to the former movement, and indicates pressure. As the deflections of the plate M are not exactly proportional to pressure, the back of the mirror is so curved at the point *l* that the necessary compensation is affected. The pipe N is connected to a three-way cock, of the usual pattern, on the engine cylinder; and the shaft Q can be connected while the engine is running.

When the speed is 1200 revs. per m. in a four-cycle motor, the indicator diagram can be viewed as one picture on account of the persistence of vision. It can then be traced on the plate. But the best way is to insert a photographic plate in a dark slide, and expose for one or two seconds. In this manner Prof.

Callendar has obtained some excellent diagrams from petrol engines, of which two are given in Fig. 1005 as samples. In addition to the diagram itself he photographed a spark on the plate which occurred simultaneously with that in the cylinder. The case B was when the spark was made just after compression, and case A when 'advanced' to such a position as to give fullest

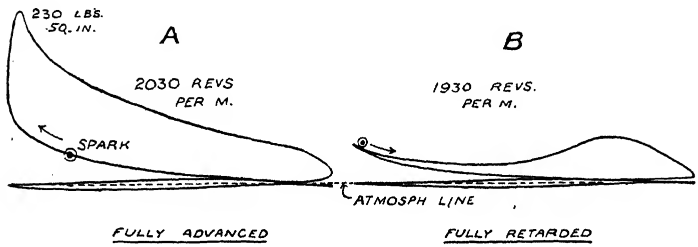


Fig. 1005. Petrol Engine Diagrams.

value to the initial pressure. Before removing the plate the atmospheric line must be drawn by suitably changing the tap of the three-way cock on the engine cylinder. The manograph is fixed upon a tripod when in use, and is placed as near as possible to the motor, so as to shorten the pressure pipe and avoid abrupt bends in the driving shaft.

Pp. 633 and 893. **High-speed Engines.**—The Belliss engine, Fig. 1006, has been considerably and successfully used for direct dynamo driving in light stations, and at one time divided the honours of that class of work almost entirely with the Willans engine, p. 893. The design of this engine is not in any way new or striking, for it is simply of compound inverted side-by-side type, as in marine practice, while the central distributing piston valve is similar to that in the Westinghouse engine, G, p. 632. Unlike the latter, the Belliss engine is double-acting, and it is also of larger size. Its success has depended very much on its careful build, but most particularly on the way in which any possible 'knock' is resisted by oil under pressure. The shock due to the reversal of reciprocating parts was formerly softened by the compression or cushioning of steam at the end of

each stroke, but when high-speed engines were introduced it was found that it was quite impracticable to supply sufficient cushion from the steam to meet the then increased forces of reversal. Two methods were introduced: Willans adopted the air cushion

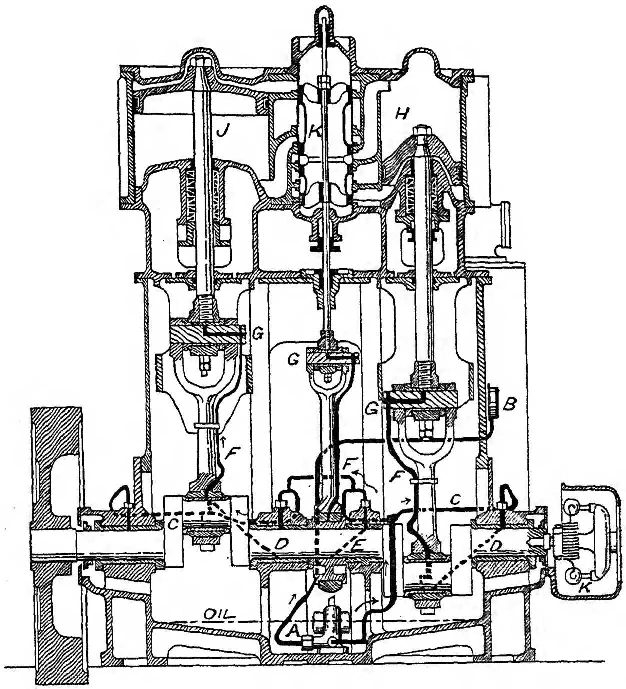


Fig. 1006. The Belliss Engine.

(H, p. 893) in a separate air cylinder, but Belliss and Morcom attacked the problem in quite a different manner, which will be understood from Fig. 1006. The bottom of the crank-shaft chamber forms an oil tank, in which is placed the little oscillating pump A, worked by a rod from the valve eccentric. Oil is

delivered under pressure to the bearing pipes C C, and then by the channels D D E in the crank shaft to the crank pins. From the crank-pin journals it travels through pipes F F F to the gudgeons, and finally to the crosshead slides, whence it returns to the tank. Two important things have been secured: (1) the oil has been fed under a pressure of 10 lbs. per sq. in. to all the moving pins and journals, so that the shock of reversal is taken by a fluid which is elastic through leakage but yet highly resistant, and (2) the engine has been efficiently lubricated. Practically any 'knock,' however severe, can be taken up by this method, though of course lack of balance is always finally transferred to the foundations and felt as vibration. The steam first enters the chamber K, and is distributed to the high-pressure cylinder H, from which it is exhausted to the low-pressure cylinder J. The governor K is of the same type as that on p. 893.

P. 640. Link Motion.

Equivalent Crank.—Let O D, Fig. 1007, be an engine crank, and O A, O B two eccentric cranks, whose radii are r and r_1 and whose advance angles are α and α_1 respectively. It is required to find a single crank that will give the same motion to C that the *combined* cranks will give. Complete the parallelogram O A B E, and the diagonal R will be the equivalent crank. For, dropping perpendiculars to A_1 B_1 E_1; $O A_1 = B_1 E_1$ and therefore the motion due to R will equal $O B_1 + O A_1$ or that due to the cranks r and r_1 taken together. If A and B be now coupled to link $a\ b$, whose mid point is c; the slide d will only receive half the travel due to equivalent crank R.

In Fig. 1008 we require a crank that will give the *relative* motions of the slides $d\ e$ as caused by the cranks r and r_1. Join A B and complete the parallelogram E B, when R will be the required crank. For $O E_1 = A_1 B_1$ which is the differential motion $O A_1 - O B_1$ excepting for change of sign, which puts O E in the left quadrant.

In Fig. 1009 two cranks are connected to a link where the valve spindle C is not centrally adjusted. C will receive $\frac{CB}{AB}$ of A's motion plus $\frac{AC}{AB}$ of B's motion. To find the equivalent

eccentric crank join $A_1 B_1$ and divide it proportionately to A C B. Then R is the crank required. For, drawing parallels $D C_1$ $E C_1$; O D is the crank that would give the proper fraction of A's motion at C, while O E acts similarly for B. Hence, by Fig. 1007, the final crank is $O C_1$.

McFarlane Gray has given another rule. Referring to Fig. 1010,

$$\text{Let } OK = \frac{BD \cdot BC}{DE \cdot 2}$$

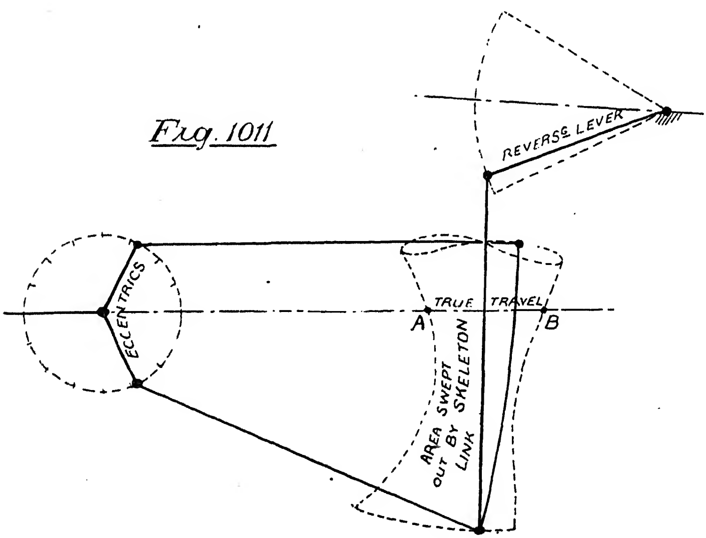
and with radius O K draw the arc B C. In B C take point K such that

$$\frac{DF}{FE} = \frac{BK}{KC}$$

Then A K is the virtual or equivalent crank. If the rods are crossed make the arc B C *convex* to A.

Correct graphic solution: The method of equivalent crank can only be used for first approximation. The proportions of the

Fig. 1011



Link Motion: Skeleton Diagram.

link motion being decided upon, strips of cardboard are provided to represent the rods and the link separately, and the actual positions of the link are set out as the cranks revolve. Properly treated, this method can be made even more accurate than a model, and a sample of the work has been given in Fig. 1011 as a notion of the process.

Pp. 650 and 773. **The Whitehead Governor,** Fig. 1012, is a most successful high-speed governor. Like the Proell governor, p. 659, p. 657, the balls move out horizontally, eliminating one resistance, that of gravity. This alone would

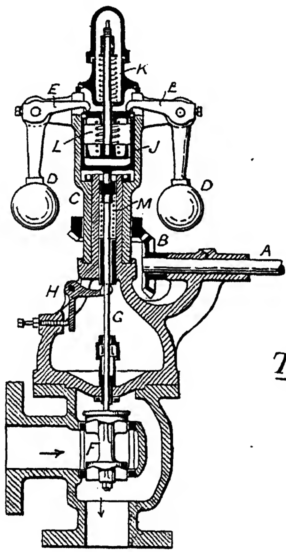


Fig. 1012

The Whitehead Governor.

make the governor over-sensitive and cause it to 'hunt,' but for the intervention of a dashpot and springs. The weights D D and arms E E form adjustable bell-crank levers that press upon the dashpot J when the balls fly out with increased speed. The piston of the dashpot meanwhile remains all but stationary, being supported by the spring K in the upper chamber. The tendency of the dashpot is to move the piston downward by pressure on spring L, and this is resisted partly by the spring K and partly by the

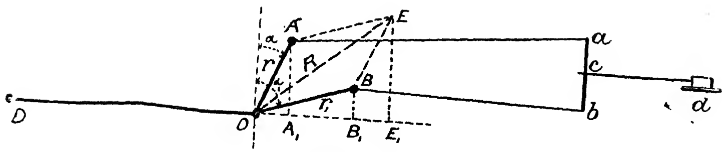


Fig. 1007

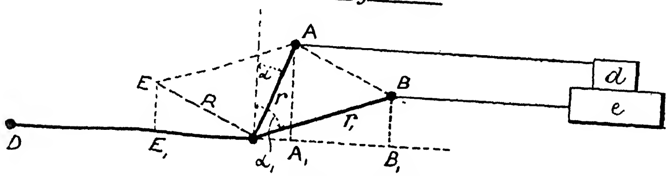


Fig. 1008

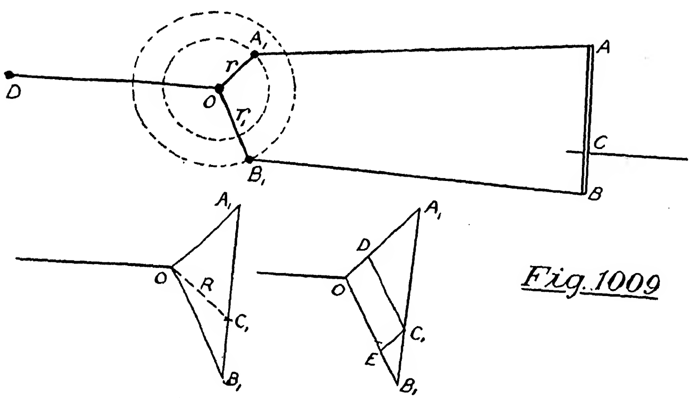


Fig. 1009

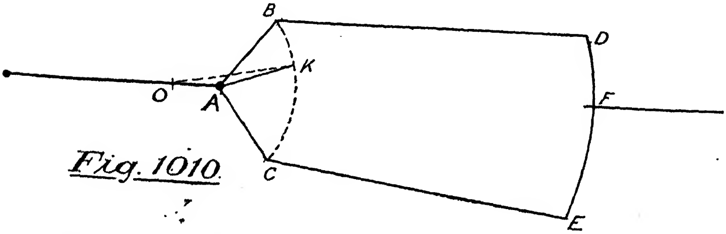


Fig. 1010

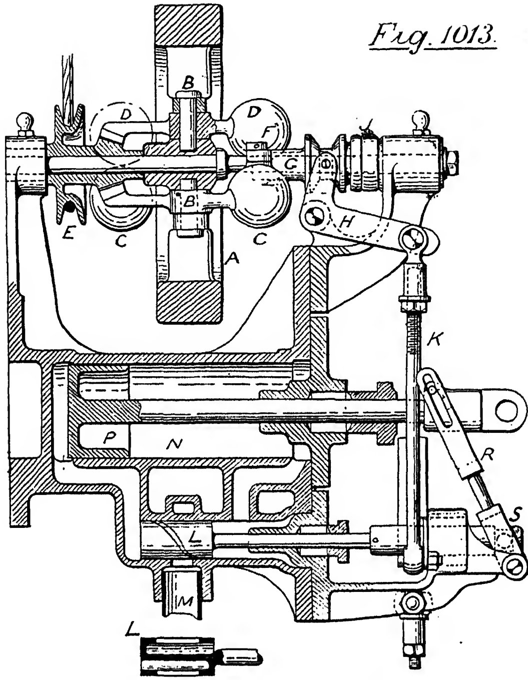
Link Motion: Equivalent Crank.

forcing of oil through a small hole in the piston. The springs, acting differentially, just balance the centrifugal force in all positions, at one standard speed. If this speed be exceeded, the dashpot is forced down against the springs K and M to partly close the throttle valve F. The piston and dashpot would remain in this new position rigidly, but the oil leaks through the piston, so as to allow the balls to fly out a little further and so throttle the steam as to bring the engine back to standard speed. The lever H can be screwed up to adjust the resistance of the spring M, and thus alter the standard speed slightly.

Marine Governors are not as much used as they ought to be, many engineers preferring to prevent racing by hand-adjustment only. That method is much to be deprecated as quite insufficient in a stormy sea, when, to adjust the steam supply with any effect at all, the engine speed must be cut down about half in the first place. There are three methods of automatic governing at present on the market: (1) where the action is obtained from the movement of the hull itself, as at p. 657; (2) where some form of balanced centrifugal governor is adopted; and (3) when a hit-and-miss arrangement is provided as on many gas engines. The last-mentioned is very little applied as yet. The second method will now be described by reference to Fig. 1013, which shows Murdoch's governor as made by Mr. John Cochrane of Barrhead.

The pulley E rotated by cord from the engine, drives the balls C C and D D, together with the flywheel A, as one solid mass; but if the speed is suddenly increased, the bevel wheel over-runs the flywheel and the balls fly outward, turning on the pins B B. The right-hand balls, being connected at F to the sleeve G, raise the ball-crank H and rod K, thus causing the valve L to rotate and admit steam to the back of piston P while exhausting from the opposite end of the cylinder N; and the eye Q, connected to the engine throttle-valve, partly closes the latter. At the same time the lever R is moved rightward, to act on spindle S and bring back the valve L laterally to the closed position. The valve is shewn below to consist of two compartments, one for admission and one for exhaust, a helical slot or port being provided in each half; and the cushion J receives the sleeve on the return of the

Fig. 1013.



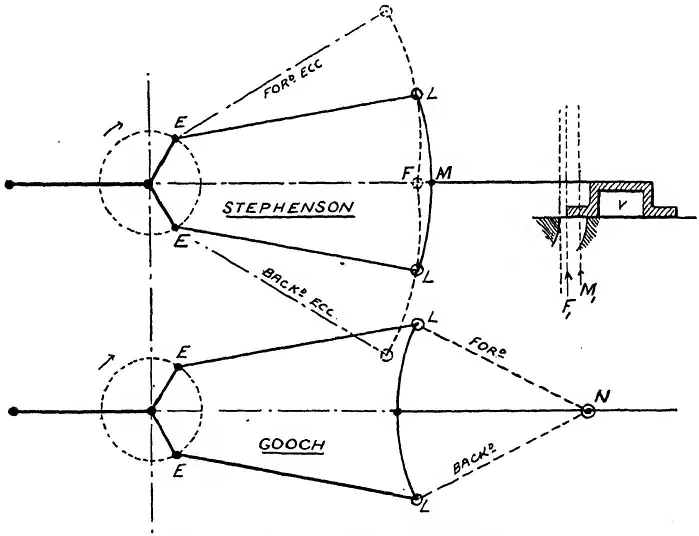
Murdoch's Marine Governor.

balls to the normal. A spring attached to H resists the tendency of the governor shaft to over-run, being tightened to any desired extent. This governor has proved very successful on some hundreds of ships, both naval and mercantile.

The Shaft Governor, Fig. 652, p. 657, as already explained, alters the expansion by a decrease of eccentric travel, and the method has one distinct advantage: the lead can be kept constant for all positions of cut-off. Referring to Fig. 1014, O C is the engine crank, and O A the eccentric crank at fullest opening. Explaining by a Zeuner diagram, the throw is decreased from O A to O B, and the cut-off position of the engine crank changes from O E to O F, when P is the dead-point of commencement. The two Zeuner circles, O A and O B, both pass through the same point M

on the line O P, and thus the lead l is unchanged. The angle of lead is, however, slightly altered from β_1 to β_2 for the admission positions of crank must pass through G and H respectively.

P. 666. **Lead in Link Motion.**—The two diagrams in Fig. 1015 represent the skeleton forms of the Stephenson and Gooch link motions when the engine crank is at dead-point. Taking the Stephenson gear, the eccentrics are at E E, and the full-gear



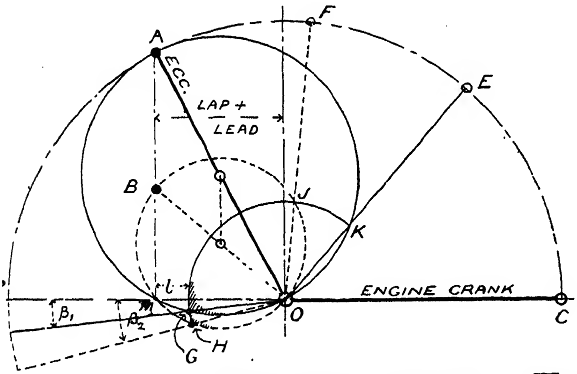
Lead in Link Motion.

Fig. 1015.

positions of the link L L are shewn dotted, bringing the valve eye to the position F, and the valve V to F, thus open to the smallest value of the lead. Moving now the link to raised position, shewn by full lines, the valve eye is displaced into the position M and the valve to M, thus opening to the largest value of lead. The linkage forms a four-bar chain rotating round the two centres E E, which explains the lead variation, the latter being found for any other position of the link by the same principle of trial. In the Gooch motion the link is never raised or lowered, and as the gear is only

anged by the movement of the valve rod L N round the centre the opening to lead is invariable. For the complete drawings these link motions see p. 640, where the Allan motion is also ewn. In that gear the alteration of lead would be only about lf that of the Stephenson, as the linking-up is only partly ected by the vertical movement of the link.

P. 686. **Edwards' Air Pump.**—The pump is used to re-ove the air and condensation water in marine engines, and is of simplest possible character. The piston A, Fig. 1016, velling within the cylinder B, has a conical under-surface that

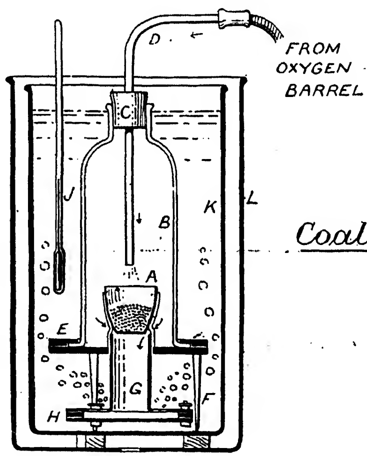


Constant Lead. Fig. 1014.

actly fits the lower cover D, and there is no foot-valve. On e up-stroke the condensed water trickles into the space D rough the opening E, while the water of the previous stroke is ted and delivered through the head valves F into the upper amber. On the down-stroke the piston reaches almost to the ttom cover, as at A_1, displacing the water, and causing it to cend through the openings C_1 into the space above the piston. is action takes place both surely and quickly on account of e free passage for air which is indicated. The improved valve ıd and guard, G and J, are fixed to the top cover, so that the tter may be lifted and the valve H examined while the pump is orking.

Pp. 693 and 910. **Serve Tubes.**—M. Sauvage finds that these boiler tubes give $2\frac{1}{4}$ times the evaporation of smooth tubes. The heating surface of a plain tube being ·92, that of a Serve tube of same size is 1·18, so the heating surface is increased nearly 30%, and the efficiency in a greater ratio. The cost of a Serve tube is $2\frac{1}{2}$ times that of a smooth one, however.

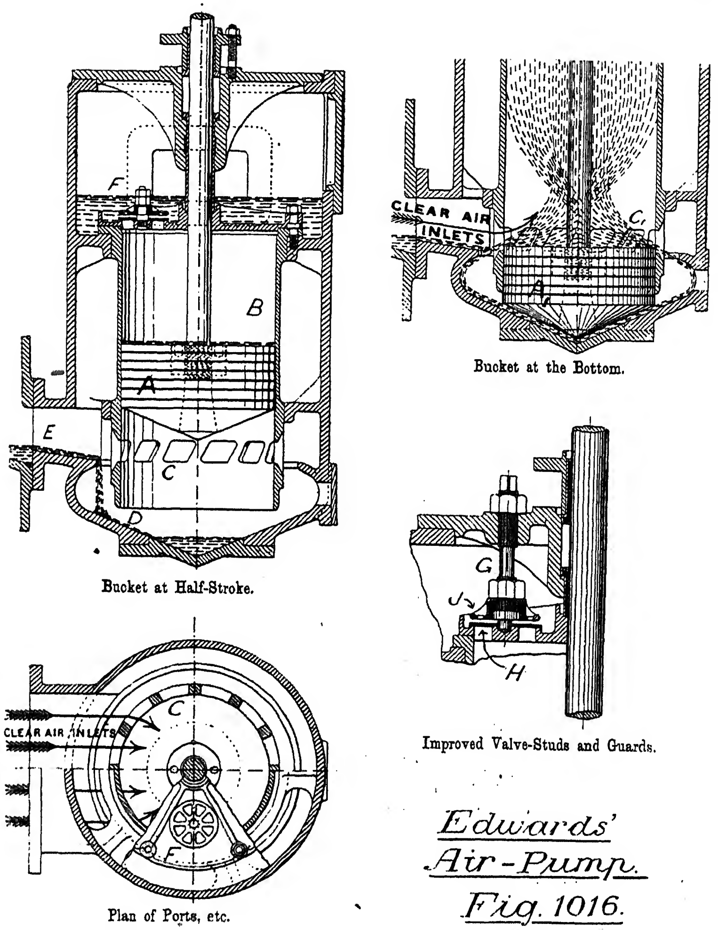
Pp. 693 and 910. **Calorific Value of Fuel.**—The *Darling Coal Calorimeter* is a great improvement on other forms, and is most perfect in action. A glass vessel K, Fig. 1017, contains the water to be heated, and is sometimes protected from air currents by being enclosed within a second jar L. The apparatus consists of a glass bell-jar B, attached to a plate E, which supports the



Darling's Coal Calorimeter.

Fig. 1017.

platinum or nickel crucible A, containing a weighed sample of the coal to be burnt. The bell-jar and stand, supported by the legs F, carry also the loose fitting G H, which consists of a tube G and a thin drum H, having holes in its upper surface for the discharge of gas-bubbles. An oxygen bottle with hand-valve adjustment is coupled to the brass pipe D, passing through the india-rubber stopper C; and a thermometer J completes the outfit. To use the calorimeter, a sample of coal is ground to powder and weighed



Bucket at Half-Stroke.

Bucket at the Bottom.

Improved Valve-Studs and Guards.

Plan of Ports, etc.

Edwards' Air-Pump. Fig. 1016.

with the crucible, the water is measured, and the combustion started by a small piece of burning sulphur. The parts of the apparatus being connected, and the whole to the oxygen bottle, they are all placed within the vessel K, and the stream of oxygen gas directed to all parts of the crucible till all the coal is burnt. During this operation the bubbles of gas from B pass upwards as indicated: delivering their heat to the water. This continuing

steadily, but without undue forcing, for 10 or 15 minutes, there is nothing left but a little ash, say between 2% and 5% of the coal weight. The rise of temperature is then noted, and a simple proportion calculation will shew the heat units per lb. of coal, or calorific value. It is best that the range of temperature should lie equally on each side of the atmospheric temperature, and thus automatically allow for radiation.

The Dowson Gas Calorimeter, Fig. 1018, is essentially a small boiler, to which the whole heat of combustion of a gas flame, as near as may be, is transferred and measured. The cylinder A encloses the chambers B and C, which are mutually connected by

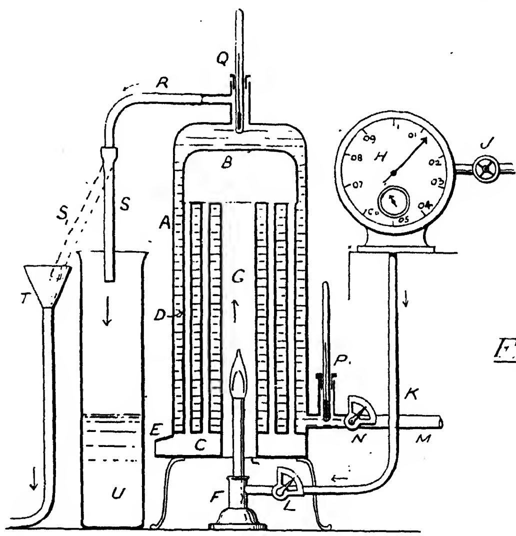


Fig 1018

Dowson's Gas Calorimeter

a number of concentric cylinders D. The gas, being lit in the burner F, its products of combustion pass up through G, and down by the cylindrical spaces to C, the condensation water emerging at E. Water entering the boiler by the pipe M, and leaving by R, is measured in the beaker jar U; while the gas burnt is measured by the meter J, to the decimals of a cubic foot. Finely adjustable

dial valves N and L, are placed on both gas and water admissions, and the thermometers P and Q shew the temperatures of the ingoing and outgoing water respectively. Water being turned on at N, and gas at J, the former is at first delivered into the funnel T, by deflecting the india-rubber tube S into the position S_1. When the apparatus is warmed and in steady working order, the test begins. Then the pipe S runs into the beaker to be there measured, the consumption of gas is noted, and the temperatures taken both at P and Q. When the beaker U is nearly full the test is stopped, and the heat units given to the water reckoned for the gas consumed, being finally reduced to the calorific value per cubic foot of gas.

Pp. 705 and 913. **Gas Engine Improvements:**

Fuel. Some remarks have already been made at p. 913 on the various kinds of gases that can be burnt with air in the gas engine. Some additional forms may be noted, giving a total choice of

1. Lighting Gas: for small and medium-sized engines.
2. Pressure-Producer Gas: } of like composition.
3. Suction-Producer Gas: }
4. Water Gas: not now made intermittently, but as at 2 and 3.
5. Mond Gas: a producer gas containing much free hydrogen.
6. Blast-furnace Gas: consisting largely of CO.
7. Natural Gas: found in the oil-bearing regions of America and the Caucasus.

Lighting Gas is too expensive for very large engines. Its principal advantage lies in the use of smaller engines for equal power as compared with poor gas, but the thermal efficiency is not increased, while the cost undoubtedly is.

Producer Gas is formed by passing steam over incandescent carbon. Formerly practised by the intermittent method described on p. 915, it was called Water Gas, but the inconvenience of the 'blow' and 'glow' action led to the invention of the continuous production of what has been sometimes called 'Semi-Water Gas,' better known in England as Dowson Gas. The same reactions no doubt take place as with true water gas, viz.:—

$$\begin{aligned} C + 2H_2O &= 2H_2 + CO_2 \\ C + CO_2 &= 2CO \\ C + H_2O &= H_2 + CO \end{aligned}$$

shewing that when steam is blown over incandescent carbon, in *due proportions*, the resulting gas consists mainly of hydrogen and carbon monoxide. Speaking correctly, the CO in the last equation is the true producer gas, and the final mixture should be called water gas. This is next burnt in the engine by the addition of 1·13 volumes of air to one of gas, the heat produced being due to the formation of CO_2 and H_2O.

Dowson and Lencauchez have both designed *Pressure Producers*, almost identical in their organs; the Lencauchez plant being described on p. 914. A small boiler is required in addition, to supply the necessary steam. *Suction Producers* are of a much simpler character, and are being built for the smaller-sized engines down to 10 H.P. The fuel is either coke or anthracite, so as to avoid tar, and the economy is much greater than with lighting gas, but not so great as with Mond or blast-furnace gases, described later. Anthracite at 22*s.* per ton will provide one B.H.P. hour for $\frac{1}{10}$*d.*, while Mond gas gives the same power for $\frac{1}{20}$*d.* Fig. 1019 is a section through Crossley's suction-producer, where A is

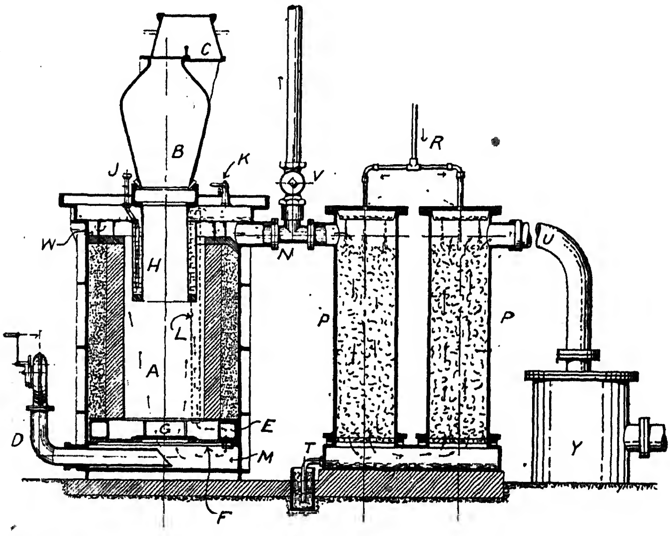


Fig. 1019. Crossley's Suction Gas Producer.

the fuel chamber lined with fire-brick and a jacket of sand, B the hopper, and C the feeder; the air-fan D being used only when starting. The fire-brick rests on a metal ring supported on two segmental boxes E called superheaters, placed on the hottest part of the fire, and mounted on a flat plate F right across the producer. This plate has a central hole across which the fire-bars G are placed. The jacket H forms a boiler to which water is fed through the pipe J; and the steam, together with air through cock K, passes down the pipe L to the chamber M, whence it rises by engine suction, and traversing the incandescent coal in A, emerges by the pipe N as unpurified gas. Passing next through the scrubbers P P, filled with loose coke down which water trickles from the pipe R, the gas is freed from ammonia, which descends as liquor into the box below, and overflows at T. Formerly a saw-dust box was placed after the scrubbers to intercept any possible tar, but the method being both unnecessary and prejudicial, has been abandoned, any tar that might form in the producer being at once converted into fixed hydrocarbons. The gas now passes to the engine along the pipe U and through the expansion box Y, the latter serving instead of the older gas-bag or equaliser. When the engine stops, the cock V is opened to exhaust the gas into the outer air: but very little is formed, as the suction action is no longer present; and the plant is easily started again, the fuel keeping hot a long time. The gas leaving the producer delivers some of its heat to the baffle plates W, which act as conductor pins to the water in the boiler.

Mond gas is obtained from common bituminous slack of very low price, say 6*s.* to 11*s.* per ton, according to district; and the process is further cheapened by the complete recovery of ammonia, as ammonia sulphate, to the extent of 6*s.* or 8*s.* per ton of fuel burnt, reducing the actual working cost to less than $\frac{1}{20}$th of a penny per B.H.P. hour. The gas is only suitable for driving engines of large size, 250 H.P. and upwards, as the plant is both large and expensive. In the smaller sizes the ammonia tower is omitted. Referring to Fig. 1020, the heating and saturating tower F is filled with tiles to distribute the hot water brought from the tank L by the pump M, which is delivered by pipe R to the top of the tower. After saturating the air from the blower E

the water falls in a cool state to the bottom at G, while the air and steam proceed to the outside jacket of the regenerator G, being thus further heated before entering the bottom of the producer A. The coal is distilled in the bell B in charges of about 9 cwt., the products uniting with the producer gas; and the heat is so great that the tar is changed to fixed hydrocarbons. In fact, the real difficulty is to keep the heat down within manageable limits, and this is effected by the supply of a great quantity of steam, 2½ tons per ton of fuel, half a ton of which is decomposed in the producer. The gas leaves the producer by the interior pipe of the regenerator C, and is next washed in the washer D by revolving splashers, taking a temperature of 194° F., and passing on to the ammonia tower H; a small portion being allowed to enter the air and steam pipe at S to still further lower the producer temperature by dilution. The towers H and K are also fitted with tiles, and supplied with cold water from G by the pump N, which delivers by the pump Q Q. Trickling through H the water becomes ammoniacal liquor in the tank J, from which it is taken to the sulphate plant, while that down the cooling tower K becomes hot at the base L on account of its meeting hot gas and receiving the latent heat of the excess steam. The cooled gas finally emerges at P on its way to the engine, but as there are some traces of tar it is filtered through a box of sawdust. The composition of Mond gas on the average is—

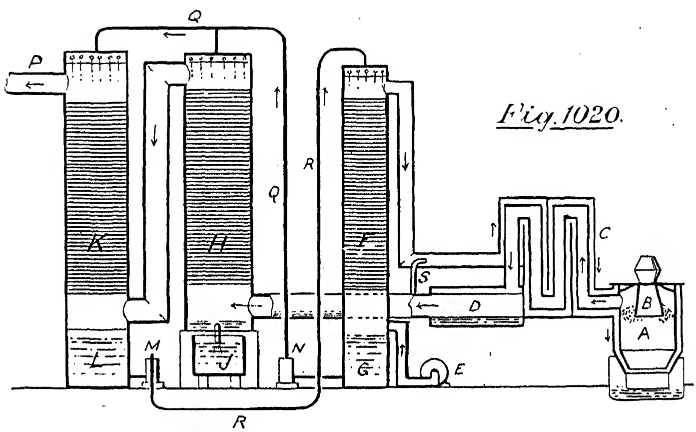
CO	11·0	
H	29·0	
CH_4	2·0	
CO_2	16·0	diluents
N	42·0	
	100·0 parts by volume.	

shewing usefully CO and a large percentage of H. One volume of gas is burnt in the engine with 1·13 volumes of air, and the calorific value is 148 B.T.U. per cub. ft.

Blast-furnace Gas was formerly and is now mainly used to evaporate water in steam boilers, the power being given to steam engines for the blowing plant. Mr. B. H. Thwaite first pointed

out that six times the power could be obtained by using the gas directly in suitably built engines, and then proceeded to carry his suggestions into effect by means of a 20 H.P. gas engine at Motherwell, when one electrical H.P. per lb. of coal was effected, as much as can be obtained only in the very best steam engines. In this case, however, the smelting was done in addition. Further, there was less difficulty in burning in a cylinder than

Fig. 1020.



The Mond Gas Plant.

under a steam boiler on account of the nature of the gas. The gas leaves the blast furnace with a fair proportion of CO, according to the reaction on p. 74, only a little CO_2 being present; in fact, an ideal producer gas. The chief trouble has been the presence of fine dust (5 %) and moisture, and these must be eliminated by suitable filtration and washing, or failure is inevitable. The composition of the gas varies according to the plant, but may be averaged as follows:—

CO	24·0	
CH_4	1·0	
H	7·0	
CO_2	9·0	} diluents.
N	59·0	}
	100·0 parts by volume.	

The calorific value is about 110 B.T.U. per cub. ft., and one volume is mixed with 1·13 of air in the engine, for which proportions the valves must be designed. The Simplex engine (p. 701) has been enlarged for blast-furnace gas, an example of 700 H.P. being shewn at the Paris Exhibition of 1900, and Fig. 1021 illustrates a complete plant for the supply and treatment of the

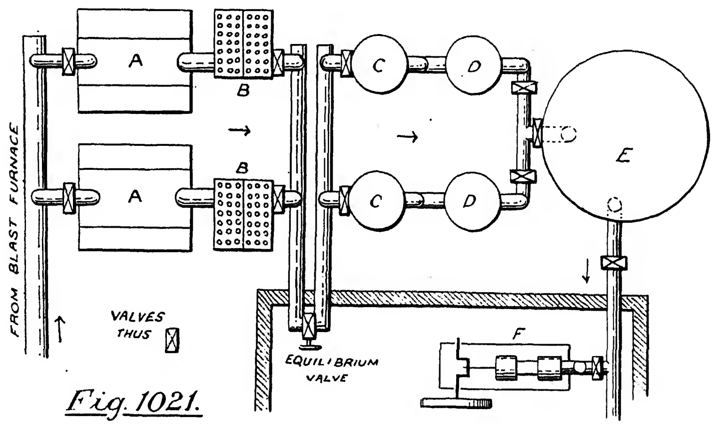


Fig. 1021.

Blast-furnace Gas Plant.

gas, where A A are washers, B B condensers to eliminate moisture, C C scrubbers to remove ammonia, and D D sawdust filters. The holder E stores the purified gas, and passes it on to the engine F.

Natural gas is obtained in petroleum districts, and its complicated composition is given by the following average:—

H	......	27·8	
CH_4	......	61·8	
C_2H_6	......	7·8	
C_2H_4	......	·5	
CO	......	·6	
CO_2	......	·5	diluents.
O	......	1·0	
		100·0	parts by volume,

shewing a rich gas with but few non-combustibles.

Thermal Efficiency of the Engine. The work obtained from coal in terms of its calorific value is much greater in any internal-combustion engine than in the steam engine, though the real improvement is a matter of recent years. The following list compares oil, gas, and steam engines :—

THERMAL EFFICIENCY.

(Ratio of B.H.P. to Calorific Value.)

ENGINE FUEL.	B.H.P.	η %
Oil	40 to 50	26 to 30
Blast-furnace gas	70 to 725	20 to 26
Producer gas	100 to 370	20 to 25
	50 to 150	14 to 20
Natural gas	40 to 600	22 to 25½
Lighting gas	10 to 50	20 to 25
Steam (simple reciprocating)	150 to 380	12 to 15
„ (triple)	1000 to 2000	16 to 17
„ (turbines)	Similar	

A series of diagrams are given in Fig. 1022, to shew the distribution of heat per cent. in various engines, and the effect of different fuels on that distribution is both curious and instructive.

The jacket and exhaust wastes appear almost to interchange, so a saving in jacket water means a hotter exhaust, while the liberal use of water, as in the Premier engine, gives the best possible result in every respect. Professor Burstall's experiments on account of the I.M.E. Research Committee are represented on the extreme right. They varied between the full and dotted line indications, the best results appearing to be due to high speed combined with small clearance space.

Improvements in the Engine. Water cooling has been carried to a greater extent than heretofore. In large engines not only

has the cylinder been water-jacketed, but water has been supplied to the pistons through jointed or telescopic pipes, flow and return, and even to the interior of the exhaust valve.

Scavenging was introduced at an early period in the Griffin and Beck engines (see p. 912), but the loss of two strokes per cycle was a serious matter, and the advantages of scavenging were therefore considered doubtful. Semi-scavenging was performed in Clerk's engine, p. 911, where the charge, prepared beforehand, was suddenly admitted, and expelled the exhaust through ample ports. The introduction of large engines with poor gas compelled the re-introduction of scavenging, when it was discovered that the effect could be obtained in two different ways, now known as positive and non-positive. The first of these has been already suggested, and the second, first used by Mr. Atkinson, consists in providing a large exhaust valve kept open longer than usual, and expelling the products into a pipe of about 65 feet length, when the wave action of the gases was such as to cause a partial vacuum of 2 lbs. below the atmosphere, and draw the whole contents away from the cylinder, while permitting the entrance of air thereto through an automatically-opening air valve. This second method, though suitable for the smaller engines, gives way to the positive method with auxiliary air pump in the larger engines. In any case, a power improvement of about 25% is obtained by the clearing process.

Greater compression has been possible with the before-mentioned improvements, without which mean pressures above 60 lbs. caused piston and valve overheating, and an invitation to seizing. Mean pressures have been gradually increased, first to 85, then to 110 and even more lbs. per sq. in.; in fact, the main improvement in thermal efficiency may be ascribed to increased compression, whose effect is shewn clearly in the following results of the Gas Engine Research Committee :—

Compression pressure in lbs.	Efficiency on B.H.P.
55	13·2
71	14·0
93	13·4
124	15·9

Premature explosion was also the risk of increased compression until better water and air cooling was introduced; being generally caused by the charge meeting the hot unexhausted products.

Governors have usually acted by the deletion of whole charges if the speed were too high, technically known as the hit-and-miss

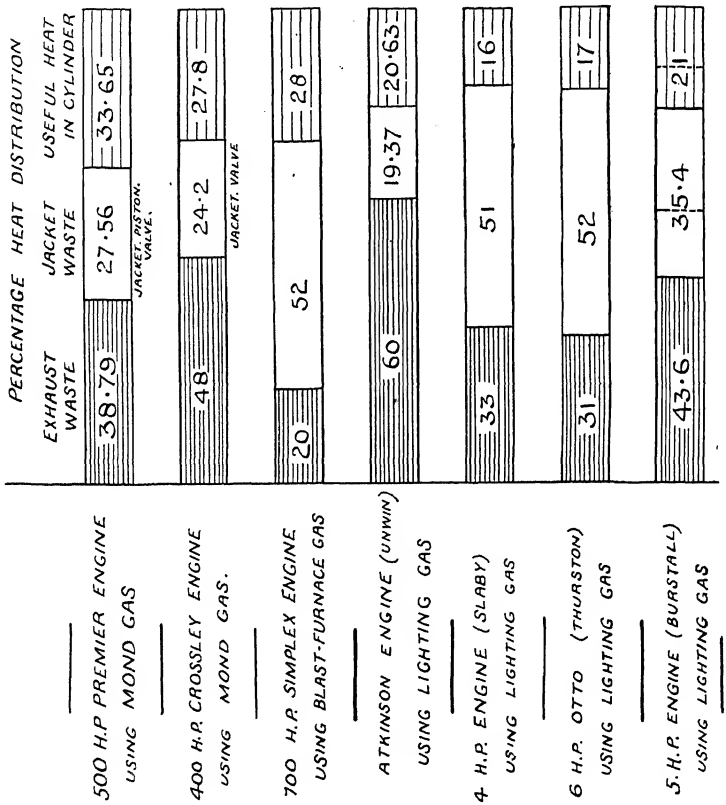


Fig. 1022. Heat Distribution in Gas Engines.

principle. Now, the governor is generally made to throttle the mixture without altering its quality, the latter being decided by the ratio of the opening of the air and gas valves. A much more

sensitive regulation ensues, permitting gas engines to be used for the exacting requirements of electric light generation.

Better-shaped clearance spaces, as indicated by the sections in Fig. 1023, have reduced the fluid resistance, which was considerable in large engines when drawing air on non-firing strokes. Nothing

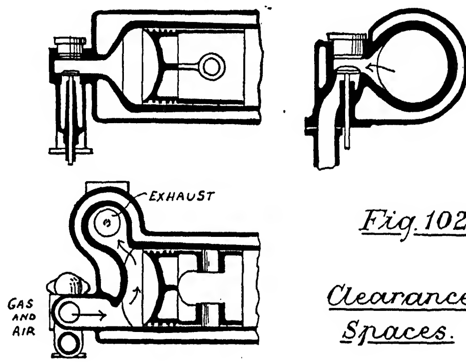


Fig. 1023.

Clearance Spaces.

but good design in pipes and clearance can decrease this. In Fig. 1024 are two indicator diagrams taken with a light indicator spring, so as to discover the area of the bottom loop: A during the firing

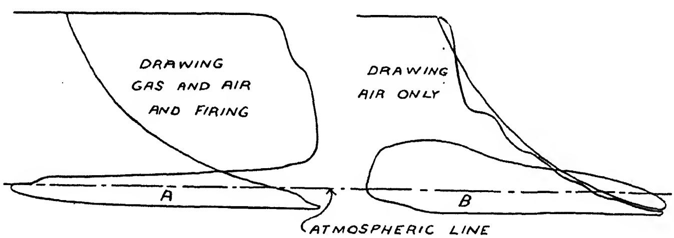


Fig. 1024. *Bottom Loop Diagrams.*

cycle and B when drawing air. The loop area represents the deduction to be made on account of fluid resistance from the rest of the diagram area, its value being about $3\frac{3}{4}$ per cent at A, and $7\frac{1}{2}$ per cent at B.

Electric ignition, pp. 700 and 963, is now almost universally adopted, being more reliable with gas than with oil engines, especially when scavenging is adopted.

Modern Engine Examples. The 500 H.P. Premier Engine, Fig. 1025, may be taken as a good example of the larger kind. There are two pistons A and B connected up in tandem to one cross-head C, which forms also a piston for air compression; and while A is coupled direct, B is driven by side rods, stretching from C to J, outside the cylinder. Air is admitted by the gridiron valve F, and compressed by the piston C into the passage H, from which it is drawn, both for charge mixture and scavenging, through the valve chambers D D. The exhaust valves E E, and the pistons A B, are all cooled internally. The valve chambers are shewn in the enlarged section, N being the main admission valve for the mixture, spring closed and lever opened at P, and R a combined air and gas valve, whose opening is regulated by the governor. When the latter valve is opened to gas, the air ports are partially closed, for the full air opening is only required on the scavenging stroke, and the mixture is so controlled as to fire easily at all conditions of load. The explosions occur in each cylinder alternately, or one per revolution of the crank.

The Westinghouse Engine, Figs. 1026 and 1027, is built in units of two or three (up to 650 H.P. and over), working on the same crank shaft, the latter form being illustrated. This method of driving produces a more regular turning moment, and makes the engine specially suited for lighting stations. The stress variations are, however, large, and necessitate a much stronger shaft than usual. Fig. 1026 is a section through one of the cylinders, shewing a well-designed connecting rod and balanced crank, and a cooled cylinder head. The half-speed shaft R is provided with a cam to lift the lever Q, which raises the rod H against a spring N, and opens the exhaust valve L, allowing the products to escape by the exhaust pipe K. Another shaft D, also rotating at half-speed, opens the inlet valve E, by the cam lever G, while the gas and air enter the mixing chamber M through the pipes A and B respectively. The mixture is throttled by the governor to suit varying loads, by means of the lever C; which raises the throttle valve within M; and the ignition is obtained

electrically at the firing plug F by the mechanical connection and separation of small levers within the cylinder. Referring to Fig. 1027, the vertical governor shaft *e* is driven by speed gear from the crank shaft, and by the levers at *d*. When the governor balls rise on increased speed a sleeve is lifted, which rotates the shaft *f* through a small angle and throttles the mixture in the chamber *h* by the lever *j*, without altering the proportions of gas and air, as before explained. These proportions can be varied, however, by the hand lever *g*, being indicated by the pointer *k*. The shaft *e* further drives the shaft *a* by 2 : 1 gear at *b*, for the purpose of

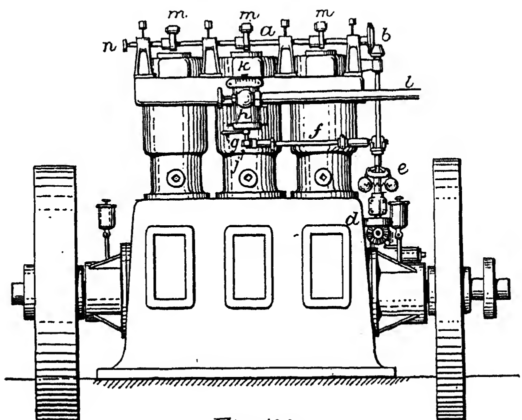
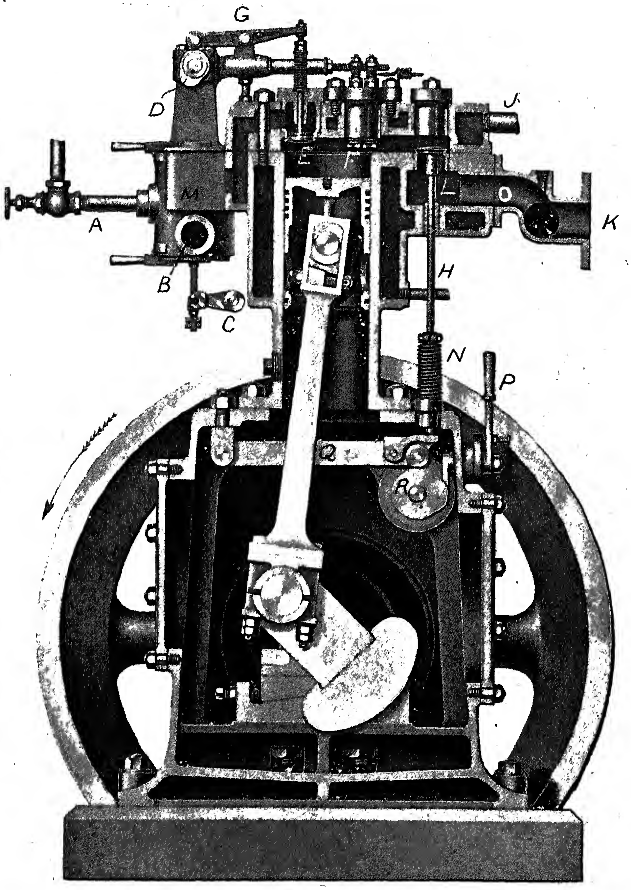


Fig. 1027.

The Westinghouse Gas Engine.

raising the inlet levers *m m*. The starting is effected very simply by the admission of compressed air from a bottle previously charged by the engine power. To do this, a milled nut *n* is turned till one of the inlet valves is put out of gear, while the handle P releases the exhaust cam. This method is equal to a barring round of the crank shaft, but in an easier manner, and the rotation is continued by the ignition of the charge. In this way, it is said, a 650 H.P. engine can be started with ease by only one attendant.

The Dingler Engine is shewn in section in Fig. 1028, and is specially interesting from its method of governing, which reduces



The Westinghouse Gas Engine.

Fig. 1026.

To face page 1162.

the quantity of mixture by an actual cut off of the supply, as in steam engines. The half-speed cam shaft A acts upon the levers S S to lift the exhaust valve R, and the exhaust pipe Q is water-

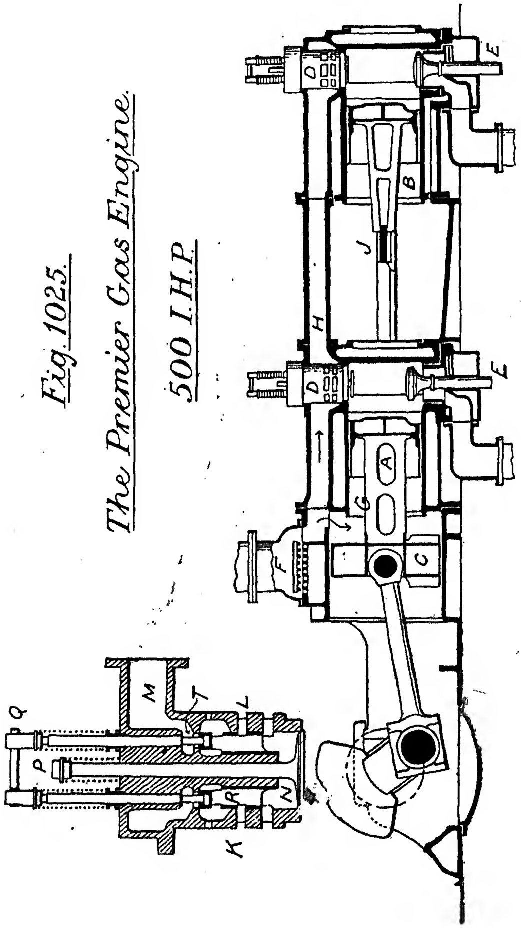


Fig. 1025.
The Premier Gas Engine.
500 I.H.P.

cooled, as well as the valve itself. The valve R admits air during the scavenging stroke. Gas enters through P and L, and air by N and M to the inlet valve L. The shaft A drives a secondary

cam shaft B at twice its speed, that is, at the crank speed. Upon this shaft is a shaft governor C connected to the cam [illegible], which is upon a long sleeve, and whose relative angular position on its shaft can therefore be slightly varied. The cam [illegible] is upon a curiously shaped lever [illegible], while the cam [illegible] presses upon a lever F F, to which is connected the roller lever [illegible] anchored at

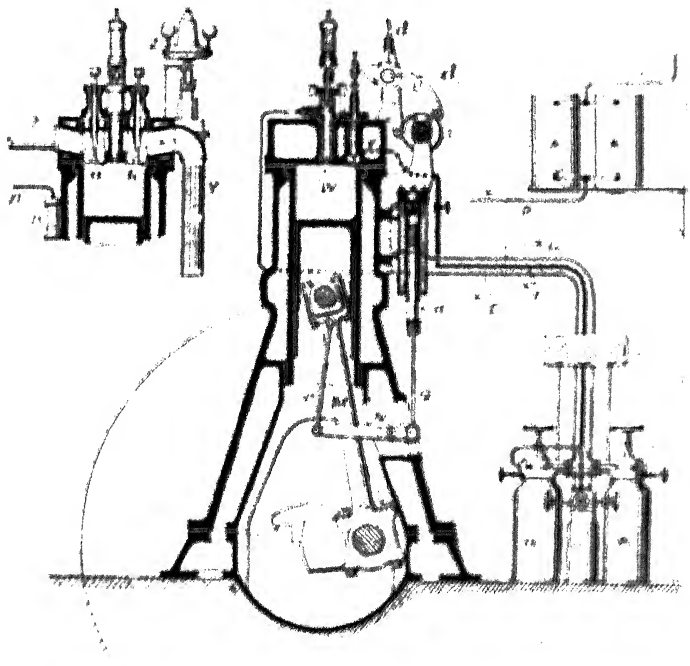


Fig 1029 The Diesel Oil Engine

H. The motions of both cams are therefore more or less apparent on the rod H, which, through lever J K, actuates the inlet valve rod. The combined cam motions *open* the valve, but the cam [illegible] only is responsible for the *closing*, and thus, as stated, will depend entirely on the relation of D and E, as decided by the shaft governor. The valve Q is for starting purposes, and the regulation of mixture is effected at L and M.

P. 709. **The Diesel Oil Engine** is a stationary engine of very high economy. Its thermal efficiency on B.H.P. has reached 28·3 %, while the oil per B.H.P. hour reduces to ·4 lb. at full load. Moreover, the cheapest form of crude petroleum can be used as fuel, the average price of which is 45*s.* per ton delivered, giving one B.H.P. hour for ·096*d.*, about half or one-third of the cost in other oil engines. As in most internal-

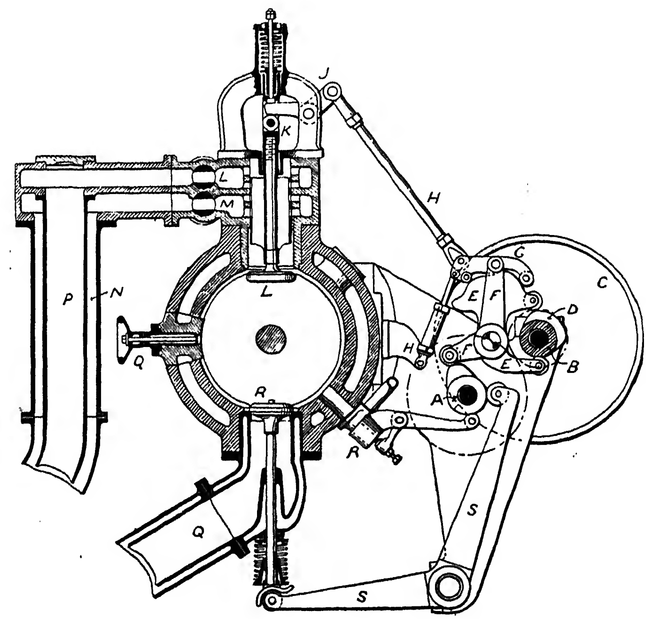


Fig. 1028. The Dingler Gas Engine.

combustion motors, the piston acts as compressor on the four-stroke cycle, and the diagrams in Fig. 1030 shew the operations. Herr Diesel set out with the intention of adopting the Carnot cycle, but was obliged to practically modify the card into (1) adiabatic compression of air from D to A; (2) admission of oil spray at A and combustion at nearly constant pressure from A to B;

(3) adiabatic expansion of the products from B to C. He also intended to dispense with water cooling, but, on account of his alteration of the cycle, was obliged to cool in a very complete manner. His proposal, however, to ignite the mixture by the compression heat has been successfully effected in practice.

Fig. 1029 shews sections through the engine as built. The air pump R, worked from the connecting rod M by the lever N and links P Q, supplies compressed air through pipe G to the reservoir J, from which again the vessels H and K are charged, H for starting purposes, and K as a reserve. The pressure in each vessel is indicated at L. In the cylinder cover are placed four different valves, of which W is a needle valve to spray oil into cylinder by air through pipe E, V a starting valve or air pump inlet at will, *b* the main air-admission valve taking through Y for the compression stroke, and *a* the exhaust valve delivering through X. These valves are controlled by levers T U, lifted by cams on the half-speed shaft S, and the oil to be sprayed is further regulated by the governor Z acting on its own throttle valve. It is most important that the oil, being coarse and unclean, should be mechanically purified by the filters A A, or the spraying needle would become clogged. These take from pipe C, and deliver through B by the action of the oil-pump D, which is worked by crank and rod from the half-speed shaft, and thus forces the oil through a fine pipe to the sprayer.

The charge of air taken into the cylinder on the down-stroke is compressed on the up-stroke to 500 lbs. per sq. in., and a temperature of 1000° F. When the piston is at the top, the spray is admitted at W, and combustion takes place during the short period of the down-stroke, indicated at A B, Fig. 1030. As nothing but air is compressed, pre-ignition is impossible, and as an excess is admitted, the combustion is much slower than is usual. The high compression is obtained by the adoption of a very small clearance, only $\frac{1}{18}$th of the working volume, and the ignition is independent of extraneous assistance. The heat, however, is considerable, and must be removed by ample water jackets, which surround the cylinder, cylinder head, and air pump.

Starting is simple and certain. Opening the test cock on the

oil sprayer and working the oil pump by hand, the engine crank is barred round till just over top dead centre. The lever D is then pulled over to D_1 which connects the starting valve and its cam. The reservoir H is next opened, and compressed air admitted to the cylinder by pipe F through valve V. The engine now rotates as an air engine. After five or six strokes, *d* is re

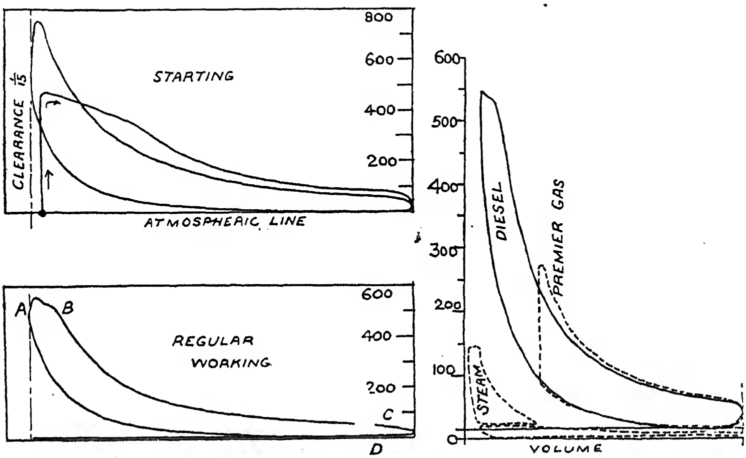


Fig. 1030. Diesel Diagrams.

placed in the vertical position, and the engine continues its course as an oil engine. Fig. 1030 shews the starting card, which clearly proves the regularity with which the engine takes its load on the oil charge. When in working order the valve V serves as a suction for the air pump, admitting thereto when the engine piston is near the end of the compression stroke. The pressure is further augmented to 750 lbs. per sq. in. before delivery to the reservoir J.

The comparison of the Diesel compression with that in steam and gas engines is made in Fig. 1030 for equal volumes of cylinder. It is this excessive compression which is without doubt the main factor in the high efficiency obtained, but the all-over

commercial economy is due also to the cheap fuel which is made possible by the cycle adopted. The engines are built in side-by-side sets of two or three units on one shaft, each piston developing usually 80 B.H.P., and each set of three 240 B.H.P.; but almost any size can be supplied from 8 to 1000 H.P. Although the mechanical efficiency decreases on light load, the thermal efficiency increases, so that the following results have been obtained on one engine at various loads:

At 160 B.H.P.	....	·40 lb. oil per B.H.P. hour.
„ 84·8 „	....	·46 „ „ „ „ „
„ 60 „	....	·50 „ „ „ „ „
„ 40 „	....	·72 „ „ „ „ „

a very creditable performance.

P. 882. **Graphic determination of $pV^n = C$.** The formula

$$\log p + \frac{AC}{CB} \log V = \log C$$

is of the form

$$y + mx = c$$

which is the equation to a straight line; where y is the ordinate, x the abscissa, and m the slope. It is therefore proved that $\log p$ is the vertical ordinate, $\log V$ the horizontal abscissa, and $\frac{AC}{CB}$ the slope of the line in the lower diagram.

P. 883. **Rankine Cycle.** Some authors refer to this as the Clausius cycle, but the nomenclature of the Institution of Civil Engineers has been adopted in this book; Fig. 848, p. 890, being called after Rankine, while Fig. 850, p. 890, is termed the common steam diagram.

Pp. 895 and 966. **The Steam Turbine.**

General considerations. This prime mover has now reached an efficiency slightly greater than that of its rival the reciprocating engine, and its study is therefore of some importance. The highest economy, it may be at once stated, is only obtained by a combination of superheating and condensation; superheats of 70° F. giving 8 % increase in power in a de Laval, and 14 % in a Curtis turbine,

or 1 % per 5° rise; while a vacuum of 28 ins. mercury saves 50 %, shewing the importance of condensation with large quantities of cold water. The best positions for turbines are therefore where water is cheap. With these precautions a consumption of 14 lbs. steam per B.H.P. hour may be easily obtained, the rate being almost constant down to loads of $\frac{1}{3}$ the maximum. Non-condensing turbines also compare favourably with non-condensing reciprocating engines, but nothing so well as condensing.

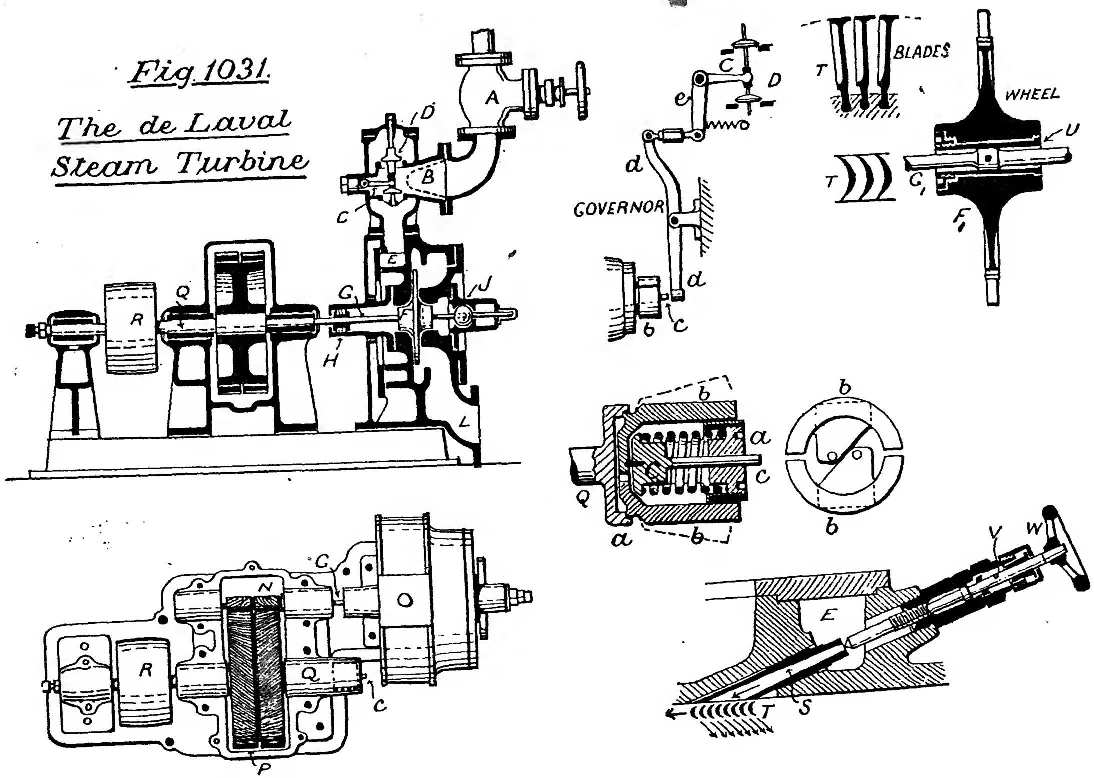
Forms. The forms of steam turbines are divided, like water turbines, into radial and parallel flow, and into

1. Single stage (one wheel) impulse turbines.
2. Multiple stage impulse turbines.
3. Single-stage reaction or pressure turbines.
4. Multiple-stage reaction turbines.
5. Partial impulse and reaction turbines.
6. Pelton-wheel turbines.

The third is represented by *Hero*, and in water by Fig. 709, p. 723. It is extremely wasteful. No. 1 has been made successful by *de Laval*, No. 2 by *Rateau* and others, No. 4 by *Parsons*, and No. 6 by *Stumpf.* The distinction between impulse and pressure wheels is not so clear when expansive fluid is used, and all steam wheels work partly in each manner. It is impossible to obtain the full impulse $\frac{v^2}{2g}$ per lb. weight of fluid, for the velocity $\frac{1}{2}v$ of the wheel periphery would then be too excessive, so all impulse wheels work partly by pressure. Generally, if the steam is allowed to acquire a high velocity before entering the wheel the impulse character is accepted, and in multiple-stage impulse turbines the steam-way should widen more rapidly than in the pressure turbines. A gradually increased pressure is necessary in both forms, and *Parsons'* and *Rateau's* turbines, though specified as pressure and impulse respectively, partake more or less of both characters, the steam giving up its energy by impulse due to change of velocity and by pressure due to reaction. The final velocity regarding wheel and the final pressure are both very low. The only purely impulse or pressure wheels are those of *de Laval* or *Hero* respectively. In the former the pressure energy of the reservoir steam is changed into velocity energy, the fluid meet-

Fig. 1031.

The de Laval Steam Turbine



ing the wheel at 1400 ft. per sec., expanding in the nozzle 1·63 times. In the reaction turbine there must be at least five wheels if a pressure of 100 lbs. is to be decreased to 15 lbs. absolute (see p. 894), and the passages should be more contracted, thus, in combination with slower speed, compelling the delivery of pressure energy. While Parsons adopted the pressure form to decrease his revolutions, de Laval accepted the speed, reducing it afterwards; and so the turbine has been developed on widely different lines.

Steam Turbine Types may be represented in classes:

Impulse Turbines.—The **de Laval** turbine is illustrated in Fig. 1031. Steam at from 50 to 200 lbs. pressure enters through the stop valve A, the gauze strainer B and the governor throttle D into the annular chamber E, reaching the wheel by three or four nozzles and exhausting at L. The nozzle S (enlarged section) is set at 20° to the wheel, and at $\frac{1}{16}''$ clearance from it. It gradually increases in internal diameter to give the 1·63 to 1 expansion, and is opened or closed by the wheel W and screwed spindle V. The steam reaches the wheel with efflux velocity of 4127 ft. per sec., and the blade centres should travel at 47 % of this or 1940 ft. per sec. if a theoretical efficiency of 88 % be desired, the leaving velocity being then 34 % of the initial, or 1360 ft. per sec. By theory this should give 11 H.P. per lb. wt., or 9 lbs. per H.P. hour. In practice the peripheral velocity is reduced to 1380 ft. per sec., thus reversing the conditions of entering and leaving, and the pressure decreases from 215 lbs. to ·93 lb. absolute into a vacuum of 28″. The wheel is mounted on a thin flexible shaft G supported at its ends in self-adjusting bearings, J being spherical and H having floating discs. On rotation the smallest lack of balance causes the wheel centre to move outward until the centrifugal force is sufficiently resisted by the shaft stiffness, when a 'critical velocity' of stability is secured and vibration ceases. This 'settling down' occurs at $\frac{1}{7}$th to $\frac{1}{8}$th of the maximum speed, and is given in revolutions as

$$N = \frac{187{\cdot}56}{\sqrt{\delta}}$$

where δ is the static deflection due to disc weight. Experimenters

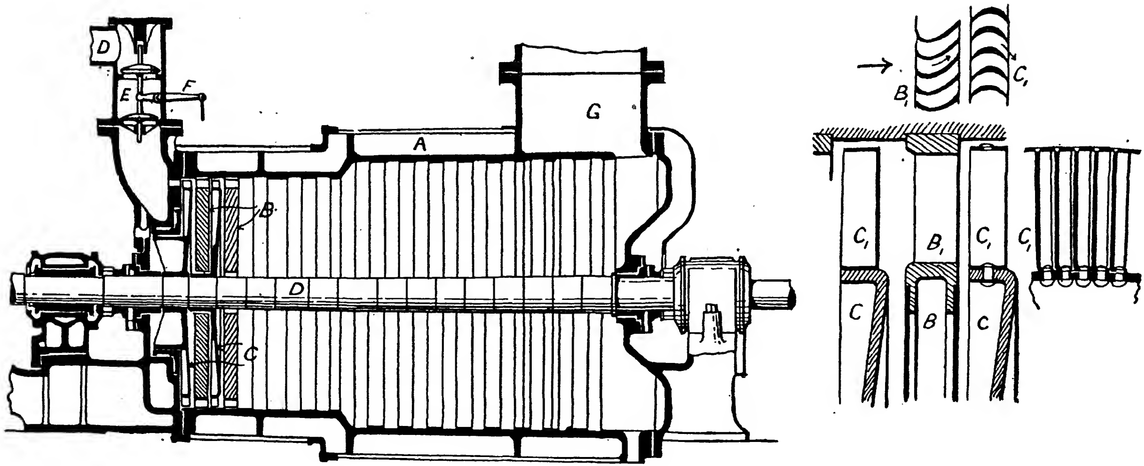


Fig. 1032. The Rateau Turbine.

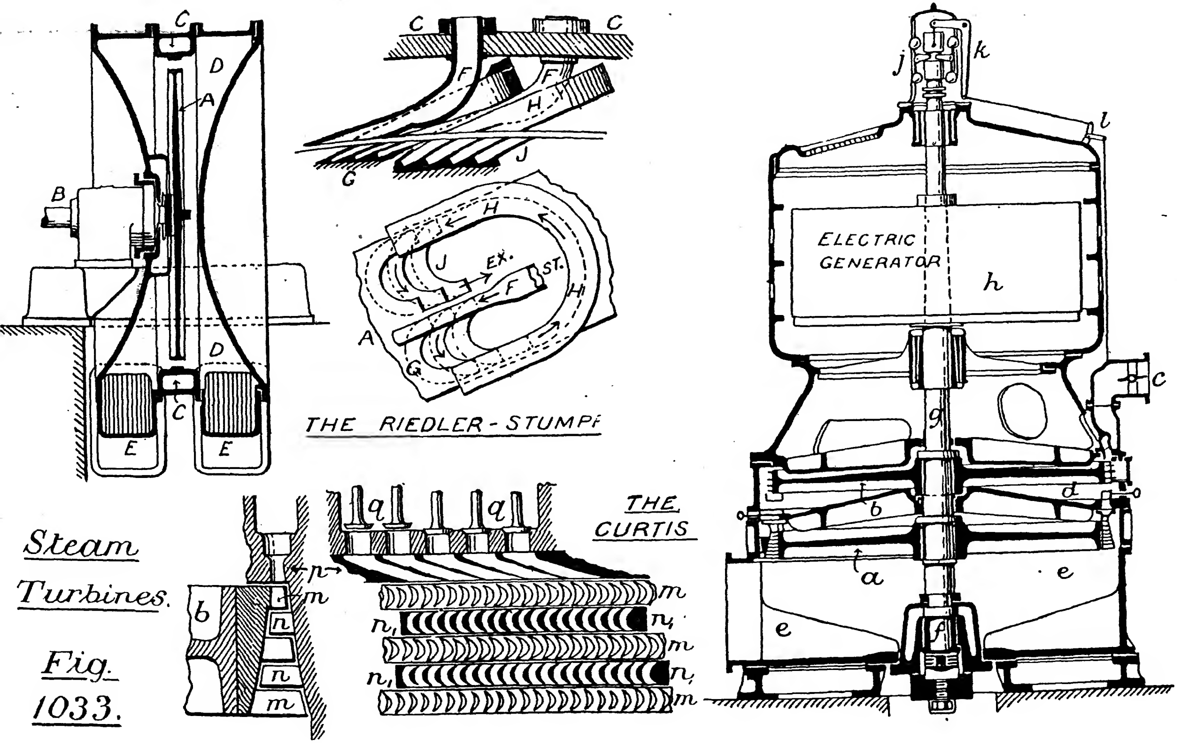
shew that the rotation is again unstable at higher velocities, and higher critical speeds have been discovered, thus :—

Critical speeds	1st	2nd	3rd	4th
Unloaded shaft	N	2N	4N	8N
Concentrated load ...	N	$2\frac{3}{8}$N	$4\frac{3}{4}$N	$10\frac{1}{4}$N
Distributed load	N	$2\frac{3}{4}$N	$5\frac{1}{2}$N	$12\frac{1}{2}$N

So, although the de Laval flexible shaft settles down at N, it would appear to reach the 3rd or 4th speed when at its maximum rotation. After once passing N the vibrations are considerably reduced between the critical points, the shaft centre line having two loops at the second, and three loops at the third speeds. Referring again to Fig. 1031, the wheel speed is now reduced by helical gearing N P, constructed in hard steel to meet the peripheral velocity of 1000 ft. per sec., and the revolutions of the second shaft Q are proportionally decreased, so as to drive the pulley R. The wheel, shewn in section at F_1, is carefully designed as to thickness so as to meet the high centrifugal stresses, which are very great, and practically forbid large wheels, or powers much beyond 300 H.P. The shaft G_1 is pinned to bush U at its centre to preserve flexibility, while the blades T T are pressed into the wheel laterally and then caulked. On the right of shaft Q is a disc and boss *a a*, on which are pivoted the governor weights *b b*, which, taking the dotted positions on increased speed, press the spindle *c c* rightward against the spring, and move lever *d d*. This acts on *e* and C to depress D and throttle the steam. The following are some general dimensions :

De Laval Turbines.

H.P.	Wheel Dia."	Revs. per m.	Rim vel. ft. per sec.	Shaft dia."
5	4	30,000	515	—
30	$8\frac{7}{8}$	20,000	774	—
100	$19\frac{3}{4}$	13,000	1115	1
300	30	10,600	1378	$1\frac{5}{16}$



THE RIEDLER-STUMPF

THE CURTIS

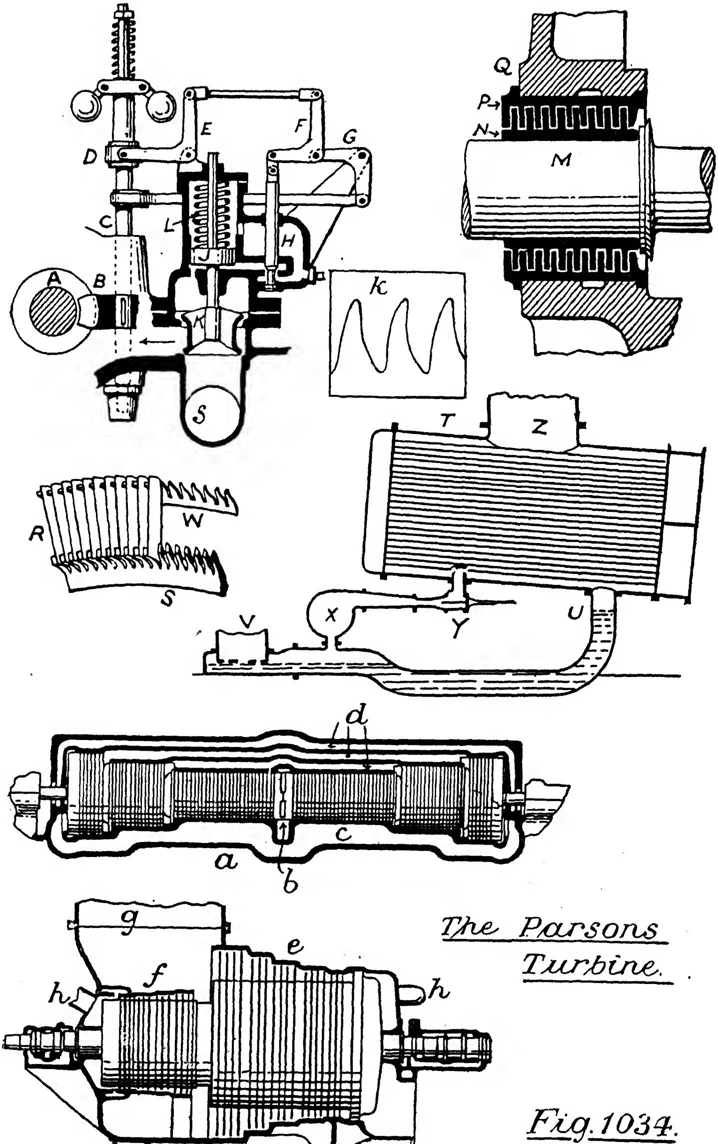
Steam Turbines.

Fig. 1033.

The **Rateau** turbine, Fig. 1032, is a multiple-stage 'de Laval' turbine. Steam enters at D, through throttle E and nozzles to wheels C C. Between each pair of wheels divisions B B are fixed in the casing A to carry guide blades $B_1 B_1$, shewn in larger section; wheels and divisions completely filling the turbine chamber. The forms of blade for wheel and casing are both shewn in plan view, and the rivet fastening is also apparent. The exhaust escapes at G. The shaft D enlarges in diameter by steps, so that the wheels may be easily driven upon it. The governor is not shewn, but can be understood from p. 721. The balls flying outward put one or other of the bevel wheels in gear on a separately driven shaft, and a screwed spindle is thereby rotated in one direction or the other to act on the throttle valve E by lever F. This turbine uses 18 lbs. steam per B.H.P. hour, the pressure falling from 120 to 1·63 lbs. absolute.

The **Riedler-Stumpf** turbine has a single Pelton wheel of most interesting construction. Referring to Fig. 1033, the enlarged section shews a wheel A with two series of buckets in parallel rows. Steam enters by nozzles F F, of which there are a large number connected to the high-pressure chamber C round the wheel circumference. The outlets have square sections in order to cover the buckets more efficiently, and the steam is rejected by the first set of buckets G G into the fixed guide channel H. The latter returns it to the wheel, where, after passing through the secondary buckets J J, it flows into the exhaust chamber at a low absolute velocity. Also the buckets J have a more abrupt angle than those at G, because meanwhile the relative velocity of the stream has been decreased, and its direction changed. The exhaust chamber D is directly connected to surface condensers E E, and the wheel A is overhung on the bearing B. A consumption of 14 lbs. steam per B.H.P. has been obtained with the wheel as shewn, as against 40 lbs. when driving one set of buckets only.

The **Curtis** turbine, Fig. 1033, has a vertical shaft *g*, driven by two wheels *a* and *b*, connected usually to a dynamo *h*, the whole being supported on a specially-designed footstep *f*. An enlarged section of wheel *b* shews three sets of blades *m m*, while



The Parsons Turbine.

Fig. 1034.

two sets of guides *n n* are fixed to the chamber. The pieces $n_1 n_1$ are of short length, existing on one side of wheel only, but sufficient to take the whole flow from the five valves *q q*. The first set of guides *p p* are placed very acutely, and have long passages, corresponding in fact to *de Laval* nozzles, and supplying velocity energy. The steam enters at C, and passes through wheel *b* to the intermediate chamber *d*, whence it traverses the blades of the second wheel *a* as it did the first, exhausting at *e e*. The governor *j* actuates levers *k l*, and by suitable rods rotates a cam shaft which opens or closes the valves *q q* in succession, so that nozzles *p p* are wholly filled or empty, thus increasing the efficiency by cut off instead of throttling. These turbines are built of 15 to 5000 H.P., and weigh ⅛th of a reciprocating engine of like power. The decrease of steam weight used when superheating is 1 % for every 5 % rise, as already stated, while in one case at least the equivalent of 1 lb. of coal per I.H.P. was expended at ¾ to full load, or equal to that of good Sulzer engines. The general consumption of steam is 16 lbs. per B.H.P. hr., the boiler steam of 118 lbs. pressure being reduced to 114 lbs. at entry.

Pressure Turbines.—The **Parsons** turbine has already been described at pp. 895 and 966 in its two forms of radial and parallel flow. As stated, the former was only adopted on account of patent difficulties, and, when these disappeared, the latter was returned to. The parallel flow turbine is, therefore, the only one to be considered, though the same principle of pressure and slow expansion is adopted in both. The turbine is shewn in section at p. 966, and a few more points will be explained by reference to Fig. 1034. The governor here shewn is that of a Brown-Boveri Parsons turbine. Steam enters at S, and flows through main valve K, which is lifted by the steam pressure beneath piston L, when valve H is closed. On the governor spindle is an eccentric C, which operates a bell-crank G to regularly raise and depress valve H, the lever H vibrating rigidly. When the latter is opened, the piston J is put in equilibrium, permitting spring L to close the main valve, and, conversely, when H is closed, K opens. The governor sleeve D is connected by levers E and F to the valve H, which it endeavours to keep raised at some certain average height,

and thus regulates the duration of steam opening. The indicator diagram is given at *k*. Parsons and others have adopted special 'labyrinth' bearings or their equivalent to allow small lateral play with steam tightness. A set of discs *w* are fixed to the shaft *w* and another set *v* to the turbine casing *q*. Slight play exists everywhere, but is taken up endwise by the thrust, and the steam is so baffled that it cannot escape. The blades *n* are fastened to the wheel *w* by bending over the saw-tooth projections and then brazing together. Sketch *a* is a double-ended turbine patented by Parsons. The live steam at *b* traverses the turbines in opposite directions to equalise and therefore eliminate thrust, the balance being further assured by the connection of each part of turbine sets through the equilibrium passages *d d*, and the exhaust is at *c*. Fig. 901, p. 969, is the usual form of turbine, however, where axial balance is procured by means of opposing pistons. Reversing turbines have been suggested and tried, where the flow of steam is itself changed in direction by alternately fixing or releasing guide or wheel blades, but they have been abandoned on account of the much lower efficiency of straight blades. In Marine work, Parsons adopts a separate turbine *f* for the reversal, while *e* serves for forward gear, the steam entering at *h h* and leaving at *g*. In some designs the turbine *f* is telescoped within *e* to save space. The benefit of a good vacuum has been mentioned, and is much greater in turbines than in reciprocating engines. The surface condenser T takes steam at *z*, and delivers as water into pipe U, both vapour and water being removed by the air-pump S. A vacuum augmentor has been introduced by Parsons, consisting of a steam jet V, using about 1½ % of that supplied to engine, which, delivering into its own condenser X, withdraws the residual vapour in T. The cold water is supplied in parallel to both condensers, and the pump does duty for all. The Parsons turbine works with 12½ lbs. of steam per B.H.P. hour, on an average, though different machines vary considerably, and 9½ lbs. of steam have been reached. The speeds are between 1500 and 4000 revs. per m.

The Schulz pressure-turbine is divided into two machines of high and low pressure respectively, and the steam traversing these

in opposite directions eliminates axial pressure, as in Parsons' turbine at *a* Fig. 1034. Also in marine work the two turbines may be so proportioned as to resist the thrust of the propeller itself.

Dynamos driven by turbines are specially designed, and 1500 revs. per m. are recommended. Marine propellers are also built of small diameter, and with a pitch suitable to the high speed of revolution.

Steam Turbine Calculations.—It is required to find the cross-sectional area of steam fluid as it expands *adiabatically* without frictional resistance. As there is no friction the expansion results in a decrease of pressure and increase of velocity such that the pressure energy of the source is entirely converted into kinetic energy when the steam pressure is zero. We shall consider the particular case of steam at 165 lbs. absolute pressure, expanding down to 1 lb. absolute, and the changes may occur either in the length of the *de Laval* nozzle, or through the wheels of a many-stage turbine, the stream areas found being the cross-sectional areas of the nozzle or blade spaces. Let suffix 1 indicate *commencement*, and suffix 2 any other position of the stream, and all calculations be per lb. wt. of steam.

$$\text{Energy at pressure } P_1 \ldots \quad \ldots = J\,(S_1 + L_1)$$
$$\text{Energy at any other pressure } P_2 = J\,(S_2 + xL_2)$$

where x is the dryness fraction.

The stream becomes wet as it parts with its internal energy, and the value of x can be found from the $\tau\phi$ diagram, Fig. 1035. Firstly, construct the diagram on a large scale by the method of pp. 887–8. Next draw line $k\,h$ at temperature 366° F. for 165 lbs. pressure, and drop perpendicular $h\,m$. As explained at Fig. 851, p. 890, the part $a\,c$ is dry steam, and $c\,b$ the condensation. A series of horizontals are drawn to represent pressures (a varying scale), and the values of $a\,b$ and $a\,c$ are shewn in columns f and g respectively by entropy units. Then—

$$\text{Dryness fraction } x = \frac{a\,c}{a\,b} = \frac{g}{f}$$

and the results are shewn on the right. Deducing from pp. 887–8, the following will represent the method algebraically :—

$$\frac{x_2\,L_2}{\tau_2} - \frac{x_1\,L_1}{\tau_1} = \log_e \frac{\tau_1}{\tau_2}$$

Putting $x_1 = 1$ for initially dry steam, and inserting all other values, x_2 may be found. The pressure energy changing into its equivalent of kinetic, we have then—

$$\text{Energy delivered up to any point} = \text{Kinetic energy received}$$

$$J\left[(S_1 + L_1) - (S_2 - x\,L_2)\right] = \frac{v^2}{2g}$$

where S_1 and S_2 are the sensible, and L_1 and L_2 the latent heats, x the dryness, and $J = 774$.

Now take a series of *pressures*, and tabulate the *kinetic energy* and the *velocity* at each pressure.

Next,

$$\text{Area} \times \text{velocity} = \text{specific volume} \times \text{dryness}$$

$$Av = Vx$$

and, $$\underline{\text{Area}} \text{ in sq. ins.} = \frac{144\,Vx}{v}$$

The values of p, v, and a are finally plotted as in Fig. 1036, on a base of kinetic energy. Two important results may be deduced: (1) the vertical np through 15 lbs. pressure divides the base into 135,000 and 115,000 foot pounds, the first of which is performed before reaching atmospheric pressure, while the

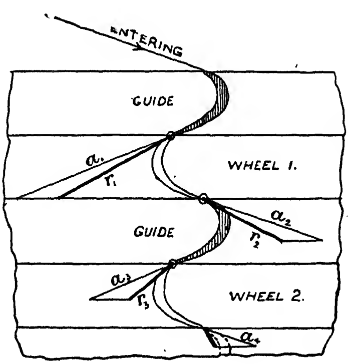


Fig.1037. Velocities.

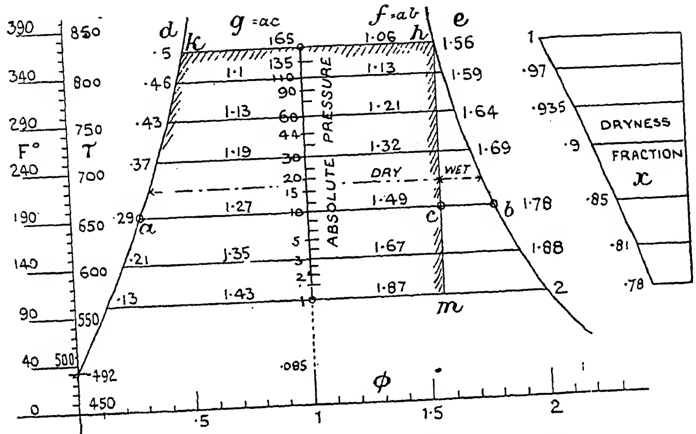


Fig. 1035. Entropy of Steam 165 lbs absolute EXPANDING ADIABATICALLY TO 1 LB. ABS.

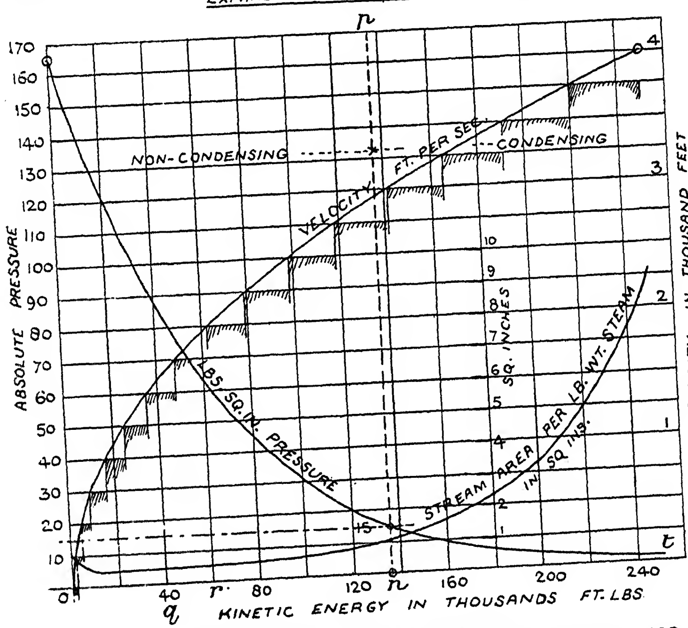


Fig. 1036. Adiabatic Steam Flow 165 TO 1 LB. ABS.

second is obtained by the use of a vacuum of 28″, shewing the importance of condensation. A *de Laval* nozzle would have a length represented by $o\,t$ if condensing, but of $o\,n$ if non-condensing. (2) Dividing the velocity curve into equal vertical drops, we shew a number of shadings that indicate the wheels of a multiple-stage turbine, when each absorbs an equal change of velocity. It will be seen that 40,000 ft. pds. require 6 wheels at first, 2 afterwards, and finally one, which means that the later wheels are more efficient, and that the length $o\,q$ should be replaced by an introductory nozzle.

The design of a steam turbine is too extensive a subject to enter on here, but Fig. 1037 may be referred to as shewing the way in which the velocities change as the fluid passes through the wheels. Taken very simply in the first case as an impulse turbine with non-expansible fluid, the steam enters through the first set of guide blades, and passes into the first wheel with an actual velocity a_1 which becomes a relative velocity of r_1 if wheel velocity is w. Entering the second guide, where a_2 is the actual velocity, equal to r_1 and placed at guide angle, this is changed to the relative velocity r_2. Again r_2 is put as a_3 to enter the second wheel, and this leaves the wheel as r_3 but at angle of a_4 to enter the next set of guides: and so on. The limit is reached when $a_4 = w$. To be correct, however, this construction must be used in conjunction with the velocities as shewn in Fig. 1036, which increase as the fluid expands, and more wheels would therefore be required than are suggested. In a reaction turbine a_2 is greater than r_1 and a_3 than r_2 in a relation depending on the extent to which the passages are choked. The energy taken by the wheels will be proportional to $a_1^2 - a_4^2$.

Fluid friction is very great at the high speeds involved, and this will modify the calculations considerably.

Pp. 709, 915, and 999. **Petrol Motor Cars.** — The general form of the *chassis* of a modern car has settled down, and extraneous types are gradually disappearing. The latter is unfortunate where such types are clever and well-working, but the demands of economical manufacture make some sort of standardisation desirable. The Live-axle Car has developed simultaneously with the Chain-driven Car, neither being now belonging to

any particular size only. In the former universal joints, and in the latter chain-driving are both objectionable, though necessary to meet the movement of the back springs.

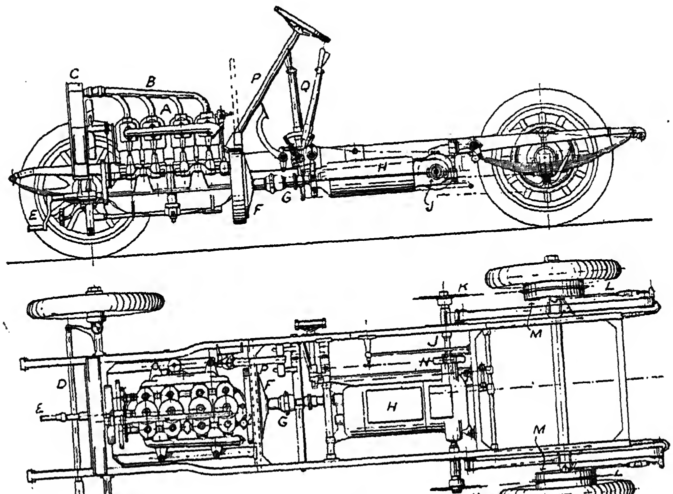


Fig. 1038.

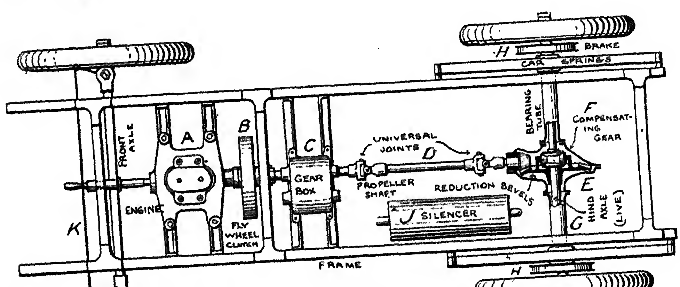


Fig. 1039.

Motor Cars.

The Chain-driven Car has been described at p. 1001, and is further illustrated in Fig. 1038, where engine frame is shorter, and there are as before four cylinders A, with a water-pipe B delivering

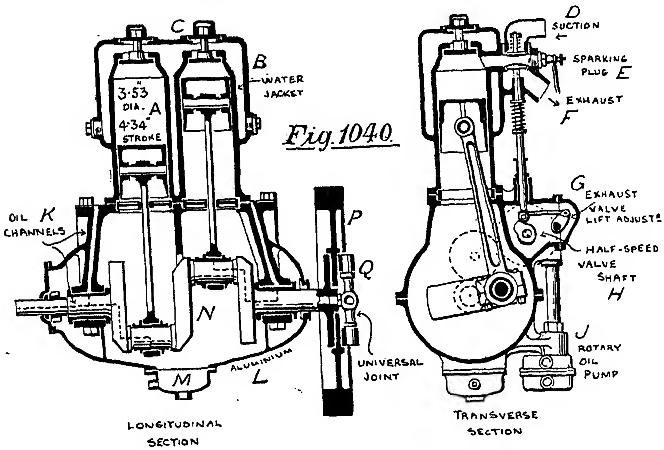


Fig. 1040.

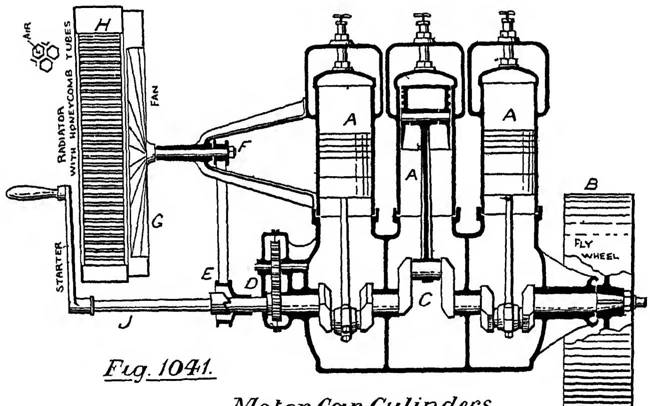


Fig. 1041.

Motor-Car Cylinders.

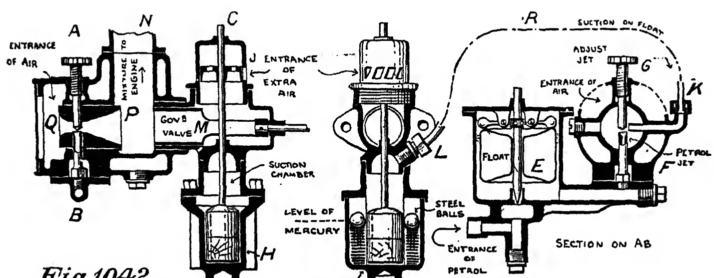


Fig. 1042.

Carburettor.

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to the honeycomb radiator C in front of the bonnet. The engine is started by handle E, and the fly-wheel F connects it to the 'propeller' shaft G, the revolutions being decreased in the gear-box H, and transmitted by mitre gear to the chain shaft J, from which the road wheels are driven by chains K K. The foot brake is shewn at N, and the main hand brakes at M M; and the fly-wheel clutch is released by a second pedal. Hand levers Q are for gear changing and back brake, while P is the steering pillar.

Fig. 1039 shews a Live axle Car of smaller power. A double-cylinder engine A, supported from the frame by cross girders, turns a fly-wheel B and propeller shaft D as usual, the speed being reduced by spur gear at C and by bevels at E in transmitting the rotations to the live axle. Spring play is permitted by the two Hooke's joints, and the compensating bevels F allow a differential motion between the two parts of the hind axle when steering round corners. Brakes are supplied at H, and a silencer J for the exhaust gases.

Passing to the consideration of details, Fig. 1040 gives two sections through the cylinders of a 10 H.P. double cylinder *de Dion* engine (see rule p. 1002). The cylinders A are cast with jacket B, and plugs C fill the vent holes. The whole is mounted on an aluminium casing L, in which crank N rotates, the latter being fed with oil from pump J, and carrying fly-wheel P outside. The clutch in this engine is located with the gear wheels. The exhaust cam is revolved by the half-speed or 'lay' shaft H, and the valve lift adjusted at G. The inlet D is lifted automatically, though many inlets are now opened by levers and cams, and the oil and water are drained at M. The *Cottereau* engine, Fig. 1041, is a type of a three-cylinder engine. Crank efforts are more uniform, and the masses are balanced, but not the explosions, though practically this is scarcely evident. Inlet and exhaust valves are on opposite sides, lever-lifted from separate lay shafts rotated from wheels D. The starting is effected by handle J, and pullies E F drive an air fan G for cooling the radiator H.

Of Carburettors there are two main kinds, hand adjusted and automatic, in great variety: the first being illustrated by the *Longuemare*, Fig. 920, p. 1005, and the second by the ingenious *Crossley* carburettor, Fig. 1042. Petrol enters float chamber E,

where it is kept at such level as to just ooze at nipple F, its outlet being further regulated by handscrew G. A wooden float H dips in mercury in chamber D, and carries by a vertical rod, the piston valve J controlling the admission of extra or adjusting air. A pipe R connects the second nipple K with the opening L just over the wooden float; and the tubular valve M, operated from a governor, acts as a throttle on the prepared mixture on its way to the engine. When the carburettor is in action, the petrol is drawn from Q to P past the spray nipple by air entering through gauze gratings; but if the suction be excessive, a partial vacuum is produced immediately over the float H by means of the suction pipe KRL, and the piston valve J automatically admits the larger amount of extra air to adjust the greater vaporisation of petrol, a mixture of constant richness resulting. The mercury is steadied by steel balls, and kept from oxidation by a film of glycerine.

Electric ignition is the only method now employed. In addition to the examples at p. 963, two other varieties are given in Figs. 1043 and 1044, where the firing, arranged for four cylinders, can be easily applied to any multiple-cylinder engine. Fig. 1044 shews the ordinary accumulator system, where the primary wire *n b* from the accumulators *a* is connected in series with the induction coil *e* and interrupting cam *c*, the former having a trembling hammer, and the latter a baseboard *d* for angular adjustment. The distributor *g* lies in the secondary circuit *m* which is connected at *r* to the firing plugs *k*, returning by earth (engine frame) at *l*, and the choice of fire is decided by the wiper in contact at *h*. Both cam and distributor are rotated by chains at equal speed with the valve shaft *f*, and four times in a revolution (depending on number of cylinders) cam *c* breaks the firing circuit, and a highly intensified current is induced, which sparks at the plugs *k k* in one cylinder after another; the usual order of cylinders being 1, 3, 2, 4, thus taking opposite cranks in pairs. The wiper baseboard *g* and trip board *d* can both be simultaneously turned through a small angle, as shewn at $d_1\, g_1$, by the lever *q*, in order to advance the spark, that is, cause it to occur earlier in the compression stroke, a useful method of hand governing within considerable limits. The switch *p* puts the apparatus in or out of gear when required.

The Sims-Bosch magneto ignition has been described on p. 965. Objection to low-tension failures in some gases on the one hand, and short circuiting by high-tension arrangements, resulted in the introduction of the Eisemann system, Fig. 1043, where an *intermediate* tension alternating current is generated by the armature N within the magnet A, and taken off by slip rings G to the primary circuit M, an interrupting cam B breaking the circuit at H. The coil C is without trembler, for the current is already alternating, and transforms the primary into the secondary current in line P, which is connected to the distributor J, whence it is taken to the firing plugs E by L L and returned by earth D. The rest will be understood from Fig. 1044.

Governors are now fitted to car engines, and, instead of letting them close the exhaust valve as formerly, they usually act on a throttle valve to adjust the supply of mixture without altering its proportions, producing a very sensitive regulation at all speeds. The *Murray* governor, Fig. 1045, is an example. The governor shaft is driven from the engine, and, as the balls fly out with increased speed, they act upon a lever B, which relieves a spring-loaded throttle valve E, and the supply of mixture is reduced. The wedge M, however, intervenes to an extent decided upon by the hand lever L, so the governor action becomes early or late, and is therefore compelled to hold up the car at any standard speed desired by the driver.

The starting of the engine is effected by the handle F, Fig. 1046, which is pressed against the spring G till the ratchets K and J engage, and thus rotate the engine shaft H. To do this easily the compression may be prevented for the first stroke or two by opening the exhaust valve, and closing it again when the explosions commence, though multiple-cylinder engines will start easily without this precaution. The ratchets are released by the spring when the pressure is removed.

To reduce the exhaust noise the gaseous stream must be split into fine jets, while providing sufficient area to avoid undue back pressure. The *Stanley* Silencer, Fig. 1047, does this satisfactorily. Exhaust gases enter at A and pass through fine holes C D E F on their way to the outer air, being further split up by baffle pins within the pipes.

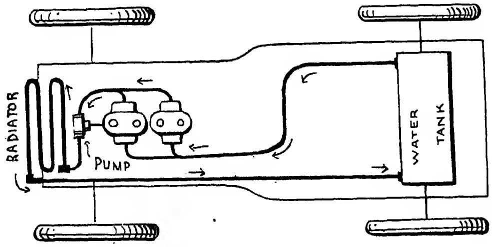


Fig. 1049

The Water Circulation.

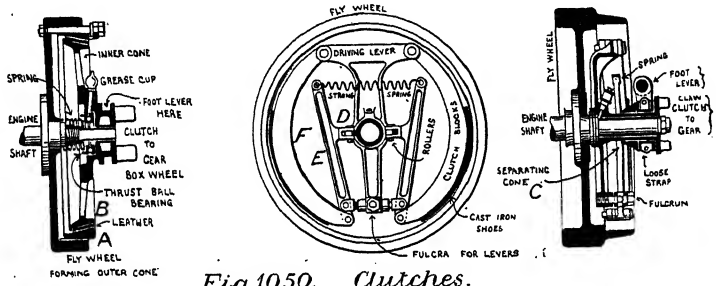


Fig. 1050. *Clutches.*

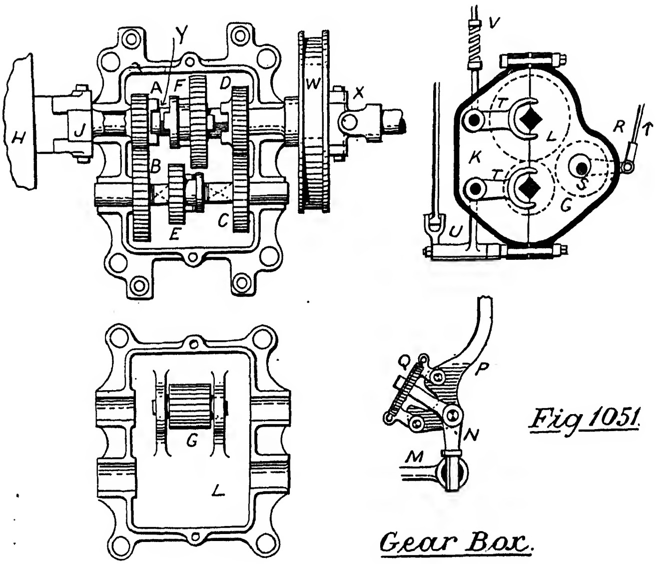
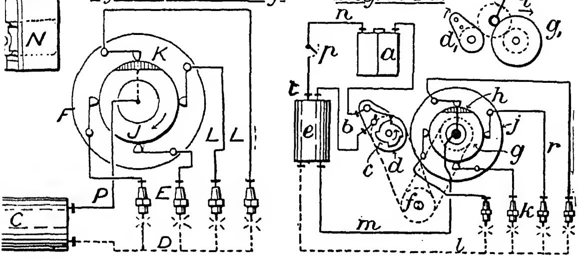


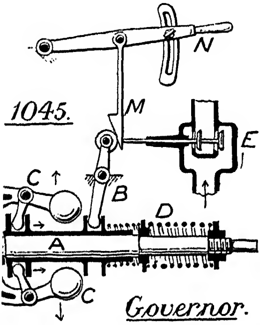
Fig 1051.

Gear Box.

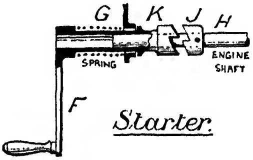
Ignition Wiring. Fig. 1044.



1045.



Governor.



Starter.

Fig. 1046.

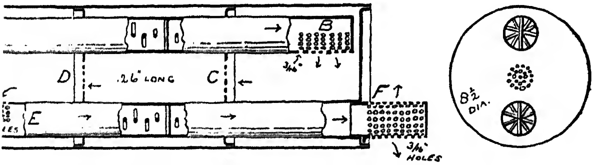
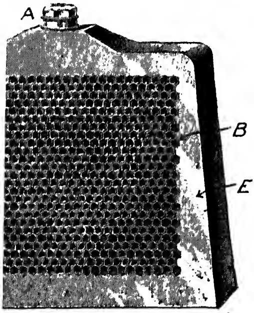
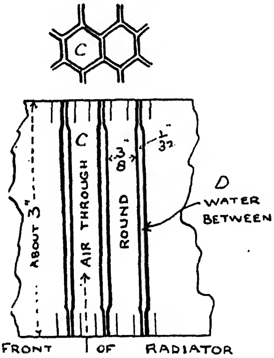


Fig. 1047. Silencer...





...tor: Fig. 1048.

The circulation of jacket-cooling water is indicated in Fig 1049, the hot water being pumped from the cylinders to the cooling radiator, then to the rear tank, returning to the cylinder to once more absorb heat. The honeycomb radiator E, Fig. 1048, is almost universally employed, where brass tubes C C of circular section are expanded into hexagons at their ends, and are soldered together by dipping the whole radiator, front and back, to a depth of $\frac{1}{4}''$ in a bath of molten tin. The space D between the tubes is only a millimetre wide, and thus divides the water stream most thoroughly when presenting it to the cooling air passed through the tubes by a fan (Fig. 1041). The cup A is for filling purposes.

Fly-wheel clutches are of two kinds. The leather-faced clutch A, Fig. 1050, has an internal spring to keep the inner and outer cones B and A in contact except when relieved by the foot lever. The leather is kept greased, and must be renewed annually. The objection to leather has introduced metal-to-metal clutches, of which the Crossley clutch, Fig. 1050, is an example. The inner portion consists of a pair of shoes F, hinged above to a central bracket, which is carried on a hollow shaft engaging with the claw clutch on first gear wheel. Two toggle levers E, connected at their lower ends to the shoes, are united at their upper ends by a strong spring which presses the shoes outwards against the fly-wheel. To release the grip, a foot lever slides the cone C leftwards, which, pressing on the rollers D, separates the levers at their upper extremities and draws the shoes together. The clutch surfaces are cylinders, not cones, and they must be kept well oiled to ensure slipping when required.

After the fly-wheel comes the gear box, Fig. 1051, an important but unfortunate necessity with all petrol engines. These are designed for four speeds on large and three speeds on smaller cars, a reversal being provided in both cases; but any number of speeds can be easily arranged by an extra pair of wheels for each. The engine and propeller shaft are disconnected at Y, and only coupled, as required, by clutching wheels A and F together. The wheel A runs solid with the fly-wheel through a hollow shaft and flexible coupling, and gears constantly with wheel B on a second-motion shaft of square section, the rotation being transferred to the propeller shaft (also of square section) by pairs of wheels, C, D

or E, F, arranged to provide the first and second speeds of 8 and 17 miles per hour; while the top speed of 25 miles is obtained by a solid drive. First it must be understood that A is locked to engine shaft: B and C are keyed to second shaft. D is fixed to propeller shaft, on which it runs freely, though coupled to X by a hollow shaft: and E F slide on square shafts. Thus for first or lowest speed E is slid into gear with F, and the drive is A, B, and E, F; for the second speed E and F are out of gear, but F is slid into clutch with D, thus driving as A, B: C, D; and the third speed is procured by clutching F with A directly. To reverse, a Marlborough wheel G is mounted on eccentric bearings S, and the latter being rotated by lever R, the drive is A B, E G, G F, which gives an opposite rotation at slowest speed. The clutch lever P is disjointed, and re-united by a strong spring Q, so the driver merely puts his lever 'hard over,' and the wheel teeth find their places by the pressure of the spring, a method less harmful to the teeth. The forks T T are intermediately operated by bell cranks U and rod V, and the whole is enclosed in casing L.

The propeller shaft is traced rearwards till it drives either a chain shaft (p. 1004) or a 'live' hind axle *t t* Fig. 1052, the speed being further reduced by chain or bevels. The difficulty of turning corners with a car of long wheel base is overcome by compensation gear *a*. Imagine a drum *a* revolving freely on its bearings, and carrying on diametral studs two bevel pinions which gear respectively with two *separate* hind axles, one to each wheel So long as the front wheels *b b* are steered ahead, the four bevels rotate solidly and the hind road wheels revolve at equal rates; but if the car be steered leftward, say, the hind wheel and bevel *e* tend to remain stationary, when *d* would revolve at double the speed of *a*, the smaller pinions serving as live rollers. In reality the outer wheel merely travels more rapidly than the inner, in automatically correct proportion (*see* p. 526). The portion of propeller shaft between gear box and hind axle is made flexible by universal joint *m* and telescopic coupling *q*, or by two universal joints. The reduction bevels *r* and compensating gear *s* are enclosed in a casing *l* which is cast with a bearing tube *k* on which rest the springs *i*, and the wheel *g* is slipped upon a

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hexagon end of the axle, so as to be easily unshipped. The casing is prevented from rotation by the steadiment *p* to frame *n*, and *h* is the brake drum.

Pitch-chain drive is shewn at Fig. 975–6, and is also men tioned on pp. 545 and 1003–5. Slackness of chain, if considerable, is remedied by the withdrawal of one or more links: if less important, by the adjustment in length of a screwed rod A, Fig. 1052, which serves as a stay between the chain shaft bracket C and the hind axle B. A right- and left-hand nut alters the length as required.

The *Ackermann* Steering Gear will be understood by reference to p. 1003 and Fig. 919. The coupling rod 4 is connected to the levers *s s*, which are splayed outward when the rod is in front, but inward when behind the axle (see Fig. 1055). These levers are again seen at H_1 Fig. 1052, and a third lever G_1 is coupled by a rod F_1 to the lever E_1, which is actuated by worm gear D_1 from the hand wheel A_1 through pillar C_1. The small levers B_1 are for advance spark, mixture, &c.

Two brakes are provided, whose general relationship is shewn at the head of Fig. 1053. The foot brake is usually placed on the propeller shaft at A, but can also be put upon the chain shaft at B, while a pair of emergency brakes E_1 E_1 are supplied on the hind axle. As brake straps (Fig. 590, p. 570) are non-reversible, they have given way to blocks, and the foot- or shaft-brake is a drum E, on the right, having two such blocks F F that are pressed together by the pedal H and lever J, the release being effected by the spring, and the torque being received on the abutment K on the frame. The wheel brake also consists of a pair of blocks F_1 F_1 all with renewable cast-iron shoes, placed within a drum attached to the wheel spokes; and the brake is put in action by the toggle lever A_1, which is pulled over by the steel cord G_1 attached to the hand lever, the upper stud B_1 being meanwhile held firmly to the frame by the rod C.

In order that the brake may be applied simultaneously to both wheels, the hand lever is fixed to a shaft lying across the frame, and the brake rods are connected to short levers also keyed thereon. An exact compensation is obtained by the Panhard steel cord *h*, which passes through the brake shaft *g* and

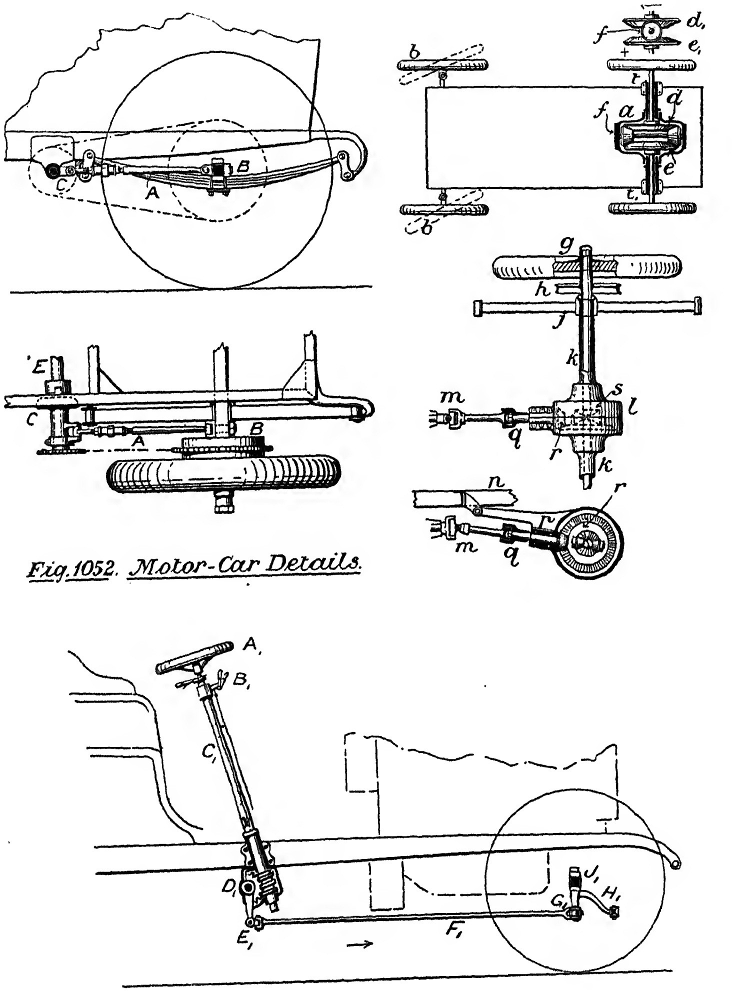


Fig. 1052. Motor-Car Details.

round pullies *j j* on the short levers. Various forms of levers are sometimes substituted for the cord, and the *Argyll* gear is seen on the left. The brake levers *b b* act upon the fulcra of bell-crank levers *e e* that connect to the brake rods *d d*, the pull being

resisted by the coupling rod *f*, which is adjustable in length. These levers, turning freely on their fulcra, are the equivalent of the Panhard rope, while being more sensitive.

As already indicated, the form of modern car has essentially settled down to that of front vertical engine and back-driving wheels, but two other forms out of the many may be described. The *Duryea* Car, Fig. 1054, is typical of American development, the engines F being horizontal, and the speed, being decreased by two sets of epicyclic gear at C, is transmitted by chain to drum E, in which are two brakes for the two parts of the live axle. The steering rod A connects to *Ackermann* levers B as usual. Each epicyclic box will give two speeds, one when the gear is in action and one when running solid, so there are four speeds and a reverse provided. Perhaps the fault of this car lies in the large portion of the running weight that is carried on the hind wheels.

The *Wilson-Pilcher* Car, Fig. 1055, is driven from four or six cylinders with opposed pistons, and any little engine vibration is further relieved by hanging the engine frame at the front end, from a high support, steadying by means of springs between the cylinder heads and the main frame. The fly-wheel L is placed in front, and the shaft clutch is at A, while the two epicyclic boxes B and C provide for speed reduction. The forward or reverse motion, as may be required, is effected by moving the wheels D and E axially as one block, through lever H, so as to gear respectively with pinion F; the compensating gear being within box G, and the brake drum at M. The change gear is shewn in Fig. 1056, where an annular spur wheel A is supposed to be fixed to the frame F, while a second loose wheel L is connected thereto through an intermediate pinion C mounted on an arm B. When B is rotated through a circle, wheel L partakes of the usual two motions, and will have revolved $1 + \frac{A}{L}$ times. If the diameter of A be twice L, the rotations of L will be three times those of the arm B. Supposing now L is on the engine shaft, and the propeller shaft is rigid with B. Then the propeller shaft will have one-third the speed of the engine shaft, and of course other ratios will give other speeds. Further, if A be freed and the arm B made rigid with L, the whole rotates solidly, and the two shafts

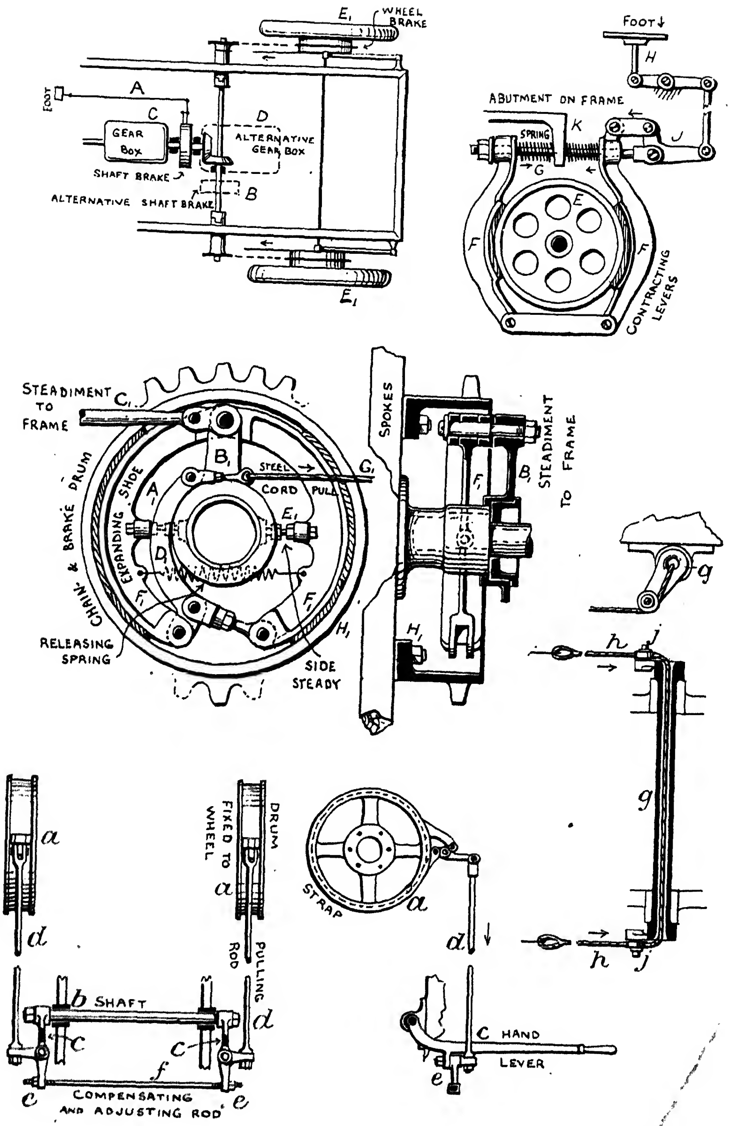


Fig. 1053. Brakes.

have equal speeds. The changes are obtained by moving A to right or left, to engage the frame cone at the low speed and the cone G for the solid drive. The two epicyclic boxes provide therefore four speeds, while the axle bevels give the reversal. Epicyclic gear has much to recommend it for cars, the teeth being always in mesh, but not in action when not reducing.

The *Gobron-Brillié*, Fig. 1057, is a balanced engine of great

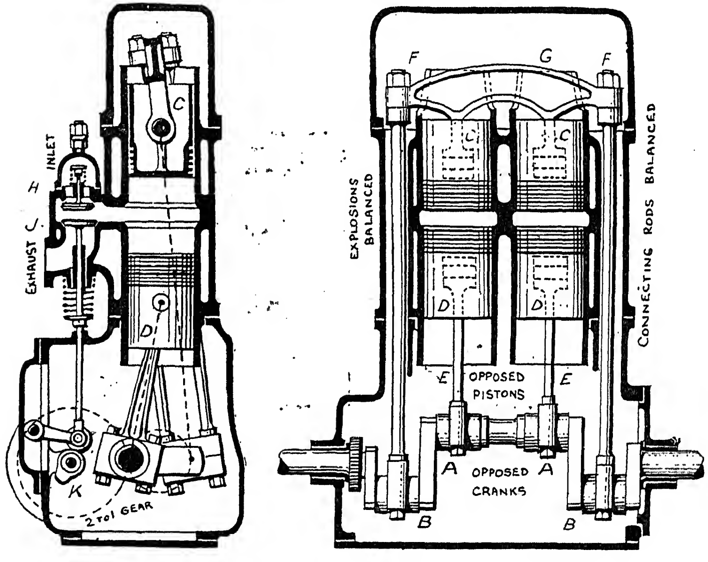


Fig. 1057.

interest. The cylinders are closed at both ends by pistons only, C C and D D, those at D D having the usual connecting rods, while the further ones C C have short rods to a transom or gudgeon G, d long rods F F outside to cranks B B which directly oppose the r cranks A A. The explosions moving the pistons simul- usly outward, the reciprocating weights in one direction are l to completely balance opposite-moving ones, and there xplosions to balance. The engines are usually arranged

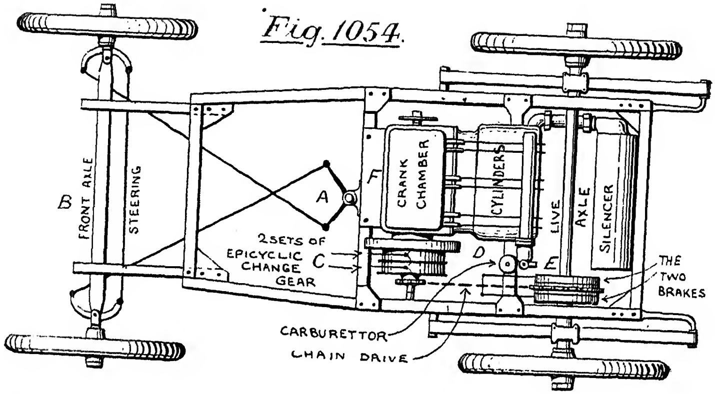


Fig. 1054.

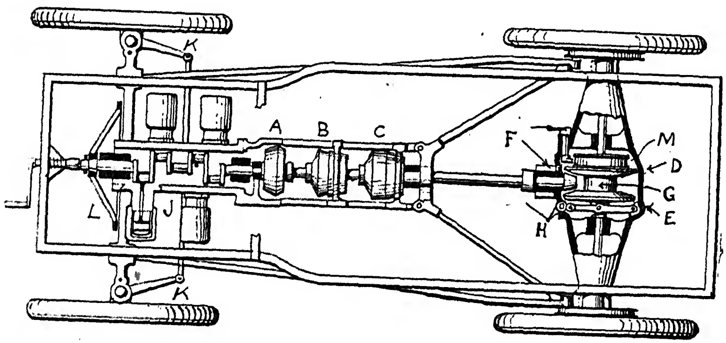


Fig. 1055.

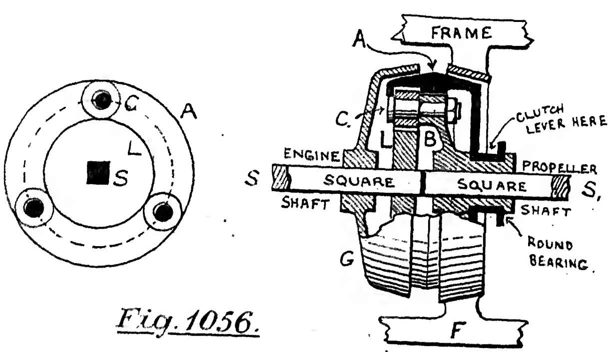


Fig. 1056.

Other Motor Cars.

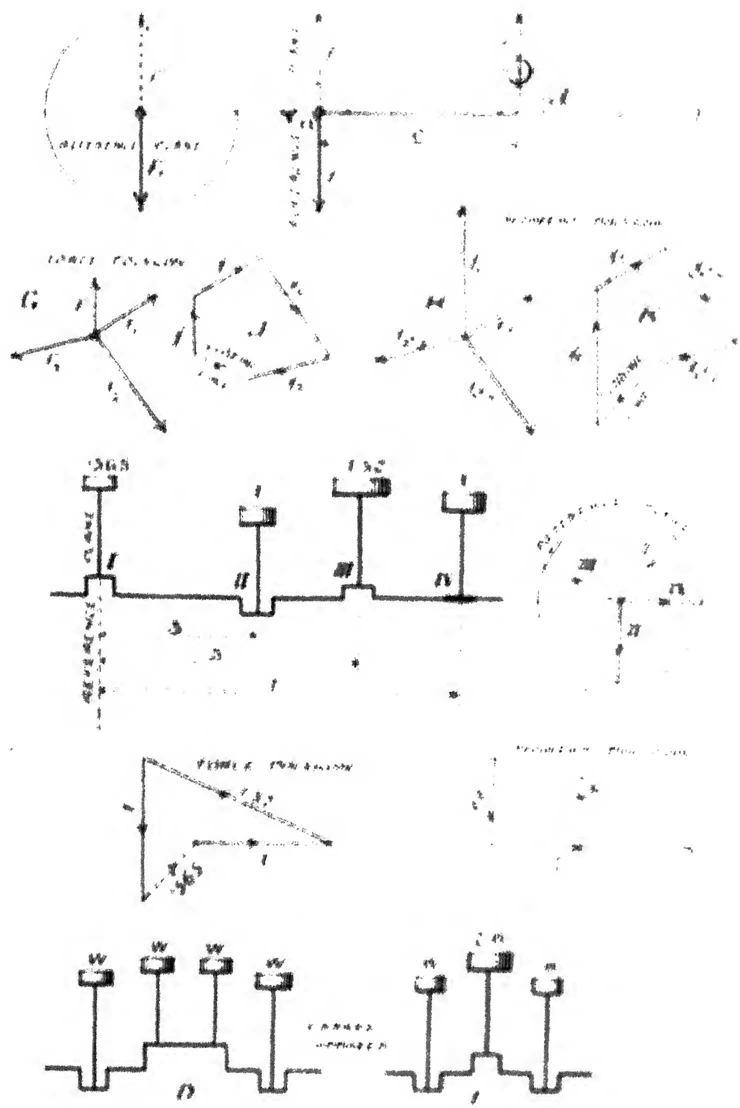


Fig 1057a Engine Balance

in pairs as shewn, but can of course be constructed with single cylinders. The half-speed shaft K and the valves H J are of the usual type.

P. 898. **Balancing.**—Professor Dalby has devised an ingenious graphic method for finding where additional masses, reciprocating or rotating, should be placed to complete the balance of an imperfectly balanced engine.

Revolving Weights.—Referring to Fig. 1057*a*, the crank shaft $a\,b$ has a centrifugal force F, due to rotation of the crank d and the weight w. Place a drawing-board or reference plane transversely at any point a in the shaft. Then, transferring the effect of F to the plane, we have

(1) A *Force* F (shewn dotted) acting at a
(2) A *Moment* F c round point a.

Calculating in this manner for any number of cranks, two sets of radial values are obtained, as at G and H, one of forces and the other of moments. For purposes of comparison, w may be substituted for its centrifugal force, and two polygons are drawn, as at J and K;

(3) A Polygon of forces f, f_1 &c.
(4) A Polygon of moments $f\,a, f_1\,a_1$ &c.

When both polygons close, there is perfect balance of the revolving weights. In the figures the closures are incomplete.

Reciprocating Weights.—If these be balanced by other reciprocating weights, the previous construction may be exactly followed: the weights being considered as collected at the crank pins; but it is quite impossible to balance reciprocating by revolving weights. Transferring their effect to a, we have, for the reciprocating weights;

(5) A Polygon of forces.
(6) A Polygon of moments.

And the polygons must both close for perfect balance.

To rectify imperfect Balance.—(7) Supply a force at a equal in magnitude and direction to the closing line in the force polygon;

(8) Place a crank and a mass in such a position on the shaft as to cause the moment shewn by the closing line of the moment polygon. As this must be done for revolving and reciprocating masses separately, we *may* have to close four polygons, and there *might* be altogether four cranks with their allied revolving and reciprocating masses respectively.

To reduce this complication, place the reference plane across the centre line of one of the original cranks, and let the angle of that crank with its mass be determined by the closing line of the force polygon for the other cranks. Also so arrange the other masses that their *moment* polygon will close. We thus have no additional crank to provide for, say, the revolving weights.

Next, let there be a definite and equal proportional relation between the masses of the reciprocating and revolving weights throughout, when each pair, though of both kinds, may be added at their respective crank-pins, and the same pair of polygons (force and moment) will serve for both sets of weights. This extreme simplification will be difficult in some cases, but the actual requirements for true balance, as they have been stated, will be a sufficient guide for any case, however stubborn; indeed, four-crank engines can always be thus simplified. It must also be clear that the motion of the reciprocating masses has here been considered as pure harmonic, and that the whipping of short connecting rods in high-speed engines would constitute an additional lack of balance.

In further explanation let us take a suppositious example. Four cranks, I, II, III, IV are placed in various positions on one shaft, II and IV having known masses = 1 in each case, while $\frac{2}{3}$ of the mass on every crank is supposed to reciprocate. Then,

$$\text{Mass moment of II} = 1 \times 3 = 3.$$
$$\text{,, \quad ,, IV} = 1 \times 7 = 7.$$

Setting these out in a moment polygon we find the closing line to measure 7·6.

$$\therefore \text{Mass moment of III} = 7·6$$

$$\text{and} \qquad \text{Mass at III} = \frac{7·6}{5} = 1·52$$

Thus the mass moments of I, II, III will balance on the

reference plane, which has been taken at centre of crank I. The force polygon is next constructed of the three forces, 1, 1·52, 1, and the closing line will shew the required direction of crank I, as well as the value of the mass (·565) to be placed on the crank. It must, however, be remembered, from our first conditions, that $\frac{2}{3}$ of this mass must reciprocate, while $\frac{1}{3}$ revolves.

Engines of accurately-balanced design have already been shewn at A and C, p. 898, and at Fig. 1057, while the method of approximate balance adopted in locomotives is also described at p. 898, where the only matter of doubt is the proportion of reciprocating weights that are to be treated as revolving. This proportion, formerly commencing at $\frac{1}{3}$ to $\frac{1}{2}$ in the older and lighter engines, has now been increased to $\frac{2}{3}$ or $\frac{3}{4}$ in the heavier modern locomotives, $\frac{2}{3}$ being the average fraction; D and E, Fig. 1057*a*, are further balanced examples.

CHAPTER XI.

P. 714. **Coefficient of Discharge.**—Fig. 1058 shews a laboratory apparatus designed to measure with accuracy the flow of water over notches, either for proving the law or for determining the *Coefficient of Discharge.* The apparatus will be here described as arranged for ascertaining the coefficient of a rectangular notch at various heads. Water flows through a 1 in. pipe into the upper tank A, being steadied by the perforated plates at D, and, flowing quietly over the notch H, arrives in the larger or measuring tank B, from which it can be drained at K. The notch plate is shewn at (6), being $1\frac{3}{4}''$ wide and nearly knife-edged. At (2) a weight G_1 carries a needle that is adjusted vertically downwards from a bracket at front of tank A, till its point is level with the lower edge of the notch. The float E_1, at (4), rests on the water in tank A and slides vertically on glass rods U U fixed to the bar S. This bar also carries a fine scale R, to shew the float level by the pointer Q, carried on a wire N. When needle J (2) has been levelled with the knife-edge, water is permitted to rise till it *just* touches the needle, and the float height is noted. This must be done with very great accuracy, so

instead of entirely adjusting by tap flow, the water level is finally raised by the displacement due to the wood block F_1 at (3), which rests on a fulcrum L and is adjusted by screw M. The fluid heads used in this experiment are very small, and unless they can be read to thousandths of an inch the results are of little value. A telescope (7) is therefore bracketed near the scale R, both original notch level and rise or head due to various discharges being noted by its help. Lastly the float V at (5), carefully calibrated, is placed vertically in tank B. It rises on guide wires W W, and its wood scale of lbs. of water is read on a fixed pointer Z, all carried on a base Y and upright *a*.

In an actual experiment under five different heads, viz. ·84, 1·08, 1·326, 1·6, and 2·123 ins., the following data and results were stated and obtained:—

1. Dimensions of notch.
2. Duration of each test (about 6 mins.)
3. Mean head at each test.
4. Total actual discharge.
5. Actual discharge per minute.
6. Theoretical discharge per minute.
7. Coefficient of discharge $\left(\frac{\text{actual}}{\text{theoretical}}\right)$.
8. Temperature of water.

The head was taken every quarter minute, and the mean obtained therefrom for each separate test. The lower tank being too small for the whole discharge, the total actual discharge was found by adding the separate fillings; and the discharge per minute deduced from the time of test. From p. 714:—

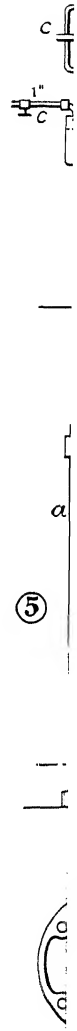
$$\text{Theoretical discharge} = \tfrac{2}{3}\,BH \times \sqrt{2g} \times H^{\frac{1}{2}}$$

$$= \tfrac{2}{3}\sqrt{2g} \times B\,(H)^{\frac{3}{2}}$$

$$= 5\tfrac{1}{3}\,BH^{\frac{3}{2}}$$

$$\text{(or putting B and H in inches)} = 40{\cdot}15\,bh^{\frac{3}{2}} \text{ lbs. per m.}$$

Tabulating the results of the five tests:—



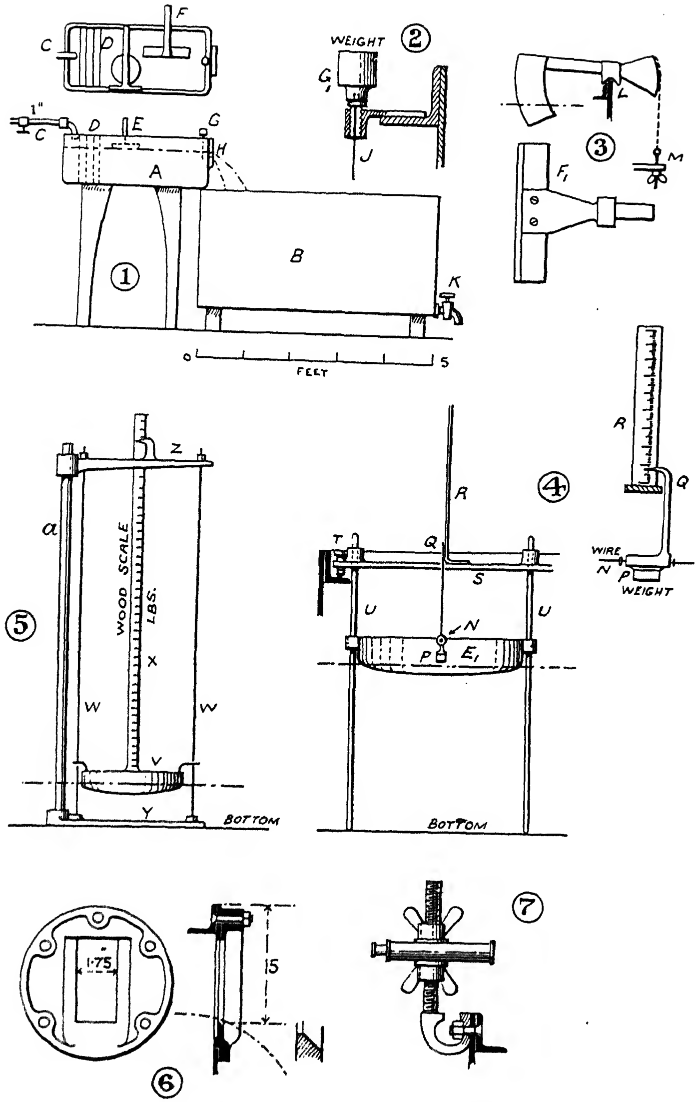


Fig. 1058. Rectangular Notch.

Head in inches.	Theoretical Discharge lbs. per m.	Actual Discharge lbs. per m.	Coefficient of Discharge.
·84	53·9	35·5	·659
1·08	78·6	49·2	·626
1·326	106·8	67·0	·627
1·60	141·6	87·8	·614
2·123	216·5	132·0	·609

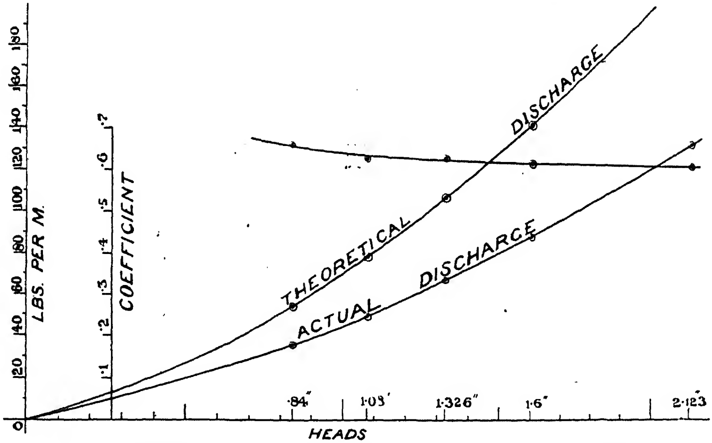
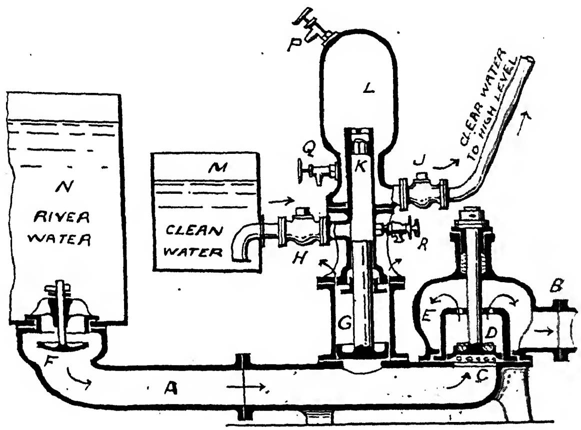


Fig. 1059. *Rectangular Notch Experiment.*

The discharges per min. and the coefficients are plotted in Fig. 1059, and shew a gradual decrease in the coefficient with larger heads.

P. 730. **The Hydraulic Impulse Ram.**—A special form of this most important contrivance is illustrated in Fig. 1060, where the power water and delivery water are obtained from separate sources. In this manner, the former may be drawn from

a more or less dirty river, while the latter may come from a clean well or spring supply. The river-water enters pipe A, and escapes in the first place through small holes in valve C: thence to chamber E, and escape pipe B. When the column of water in N and A has acquired the maximum velocity due to head, the valve C is lifted suddenly so as to strike the top of the bell D, the india-rubber block at top of valve closing the orifices in D. The out-flow of the water being stopped, its momentum forces up the



Hydraulic Impulse Ram.

Fig. 1060.

piston G, which also acts as a pump plunger, whose return stroke is obtained by its own weight. At the down stroke of G, clear water is drawn from the tank M through the suction valve H, and at the up stroke this water is forced through valve K into the air chamber L, where it finally passes through the delivery valve J to the higher level required. The supply pipe A is provided with a back-pressure valve F, which closes at the same time as C, so that the whole energy of the water is utilised, and not partly spent in forcing some of the fluid back into the tank N. The force of the water in A having been spent, valve B falls back on its seat, and G

falls also to the position in the diagram : while the pump K re-fills, and the action is repeated.

To keep the air vessel supplied with air, a snifting valve R' is fitted between suction and delivery ; and the valves P and Q serve to empty the chamber L of air and water respectively, when it is desired to stop the action of the apparatus.

With this pump 60 cubic feet of power water per minute at a fall of 3 feet has been made to lift 300 cubic feet of clear water to a height of 360 feet in 24 hours ; which corresponds to an efficiency of 41 per cent. ; but a maximum of 70 per cent. efficiency may be looked for in the case of all impulse rams under favourable circumstances.

P. 730. **Air Vessels for Pumps.**—The air in air vessels becomes gradually absorbed during the working of the pump, especially at higher pressures. It is for this reason that air vessels are not permissible in accumulator pumps (see p. 734). To re-charge such a vessel with air, two cocks may be used, one at the top and one at the bottom of the vessel, so that water may be

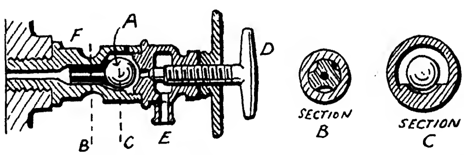


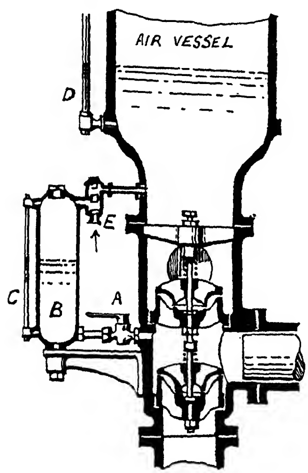
Fig. 1061.

Snifting Valve for Air Vessels.

drawn off below and air re-admitted above. If it be desired to admit it automatically while at work, a snifting valve, a separate air pump, or an 'air injector' as it is termed, may be adopted. The first of these, shewn in Fig. 1061, is placed between the suction and delivery valves ; and consists of a gun-metal chamber F, containing a ball A which rolls backward and forward on account of the pump stroke. When rolled leftward, air is drawn inward at E, while the escape of water on return stroke is prevented by a rightward roll ; and the whole may be put out of action or adjusted by the screwed plug D.

The air injector in Fig. 1062 is a much better contrivance. A

small cylindrical chamber D is connected at bottom to a point just under the delivery valve, and at top to the air vessel. The rise and fall of water in B acts as a liquid piston to draw in air at E and deliver it into the air vessel, while cock A adjusts the action. C and D are glass gauges.



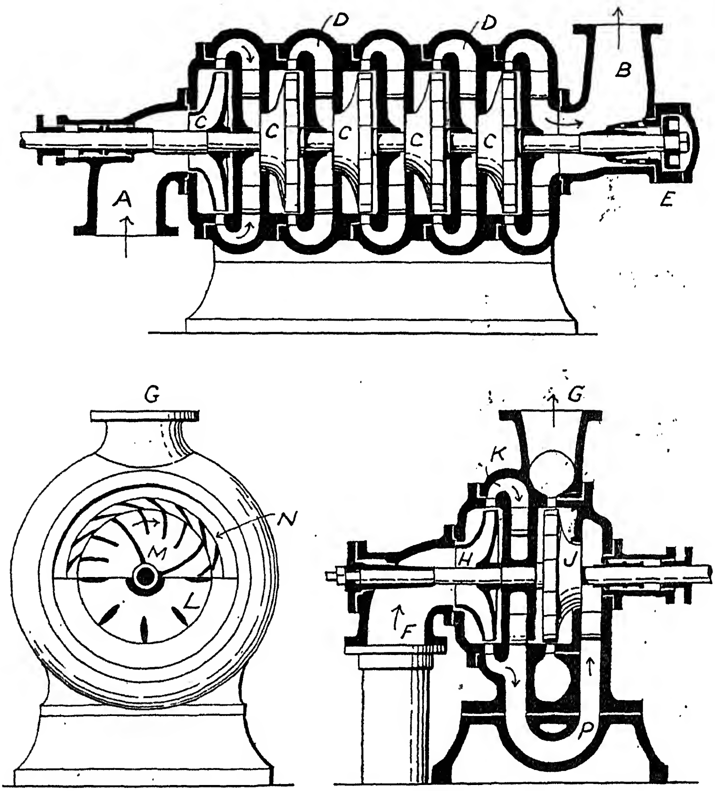
Air Injector FOR Air Vessels. (WHIPPERMAN & LEWIS)

Fig. 1062.

P. 728. **The Centrifugal Pump.**—The diagram of pump efficiencies, p. 734, shews that a single pump is unfit for high lifts because of the rapidly decreasing economy at increase of pressure. The difficulty can be overcome by compounding, that is, placing two or more pumps in series, so that the pressure-head obtained finally is the aggregate of those due to each pump.

Sulzer was the first to adopt this principle of multiple stages, and he was rapidly followed by Rateau, in Paris, and Richards and Jackson in America. Any arrangement of separate pumps having each delivery pipe connected to the suction of the next in order would fulfil the requirements, but they are more conveniently made in one casting. Referring to Fig. 1062, the upper drawing is a five-stage pump, where the main suction is at A and the delivery at B. Naturally the water enters most conveniently in an axial direction, while the wheels C C deliver at their circumferences into the intermediate passages D D; but a

leftward axial stress is thereby caused, which must be balanced by the pressure of the delivery water on the piston E. The lower

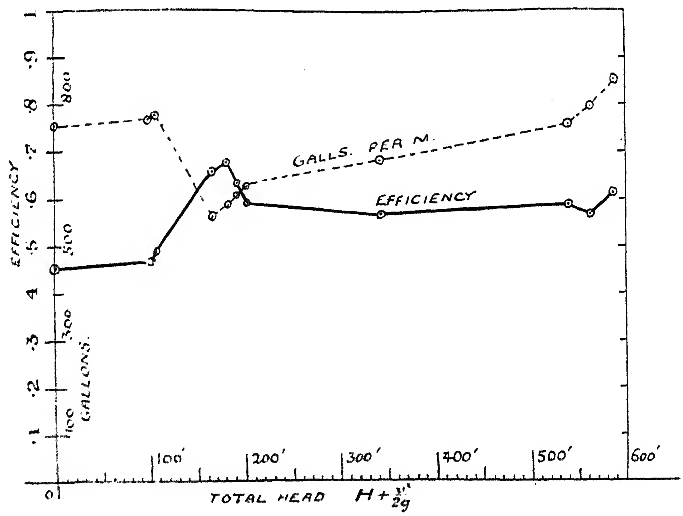


Multiple-stage Centrifugal Pumps. Fig. 1062.

drawing is a two-stage pump of a similar design, but with one great improvement: the wheels or impellers are placed back to back, and the axial stress is thereby completely eliminated. The

water enters at F, and passing the first wheel H, travels along the passages K P to wheel J, whence it escapes at G. The end view will serve for both pumps. The wheel M, rotating in the direction shewn, gives the water both tangential and radial flow. As it is desired that the fluid shall be simply pressed forward radially and quietly, the tangential velocity is next eliminated by the relative back action of the guide blades N, fixed in the casing, and having an inclination opposed to the wheel motion. The principle is simply that of a reversed water turbine, and such pumps are generally spoken of as turbine pumps. The axial balance piston was introduced by Rateau, and the back-to-back wheels by Sulzer. The latter are equally applicable to many stages, but must be so arranged that half the wheels point in the opposite direction to the other half. The guide blades are an old principle revived.

The efficiencies in compound pumps at high lifts are very good, while those on p. 734 for single wheels only reach their maximum at low lifts. As to actual work, a 20 in. six-stage pump



Efficiency of Six-stage Pump.
Fig. 1063.

would raise 20 million gallons per 24 hours to a height of 185 feet. Fig. 1063 is plotted from experiments made upon a six-stage centrifugal pump of 6″ aperture, with a wheel 22″ diameter. It will be seen that the efficiency varies with the rate of delivery in a very remarkable way, the highest values being always obtained when small quantities are being supplied at high head.

Pp. 736 and 738. Friction of Hydraulic Ram.

Example 79. A hydraulic lift with ram, load, &c., weighs 10 tons, the ram is 9″ diameter, and the friction in the mechanism is equal to ·05 of the gross load. The accumulator is half a mile away, and is loaded to 800 lbs. per sq. in., and the diameter of the supply pipe is 3 ins. Estimate the speed of ascent of the lift, if the loss of head in the pipe is $\frac{\cdot 0005\, L v_1^2}{D}$ feet where v_1 is the water velocity in pipe.

(Board of Education Applied Mechs. Exam., 3rd stage, 1904.)

Let v = speed of lift. Then, comparing areas of ram and pipe, $9v$ = velocity in supply pipe.

Putting loss of head as loss of pressure in 3″ supply pipe :

$$p = \frac{H}{2\cdot 3} = \frac{\cdot 0005\, L v_1^2}{2\cdot 3\, D} = \frac{\cdot 0005 \times 5280 \times (9v)^2}{2\cdot 3 \times 2 \times \cdot 25} = 184\, v^2 \text{ lbs. per sq. in.}$$

The pressure in ram will then be $800 - 184\, v^2$ lbs. per sq. in.

Next equate work inside ram with external work, and we have,

$$(800 - 184\, v^2) \times \frac{\pi\, 9^2}{4} \times v = 10 \times 2240 \times 1\cdot 05 \times v$$

$$\therefore\ v^2 = 2\cdot 34 \quad \text{and } \underline{v = 1\cdot 53 \text{ ft. per m.}}$$

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